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Heating · Piping and Air Conditioning

OCT 7 1929

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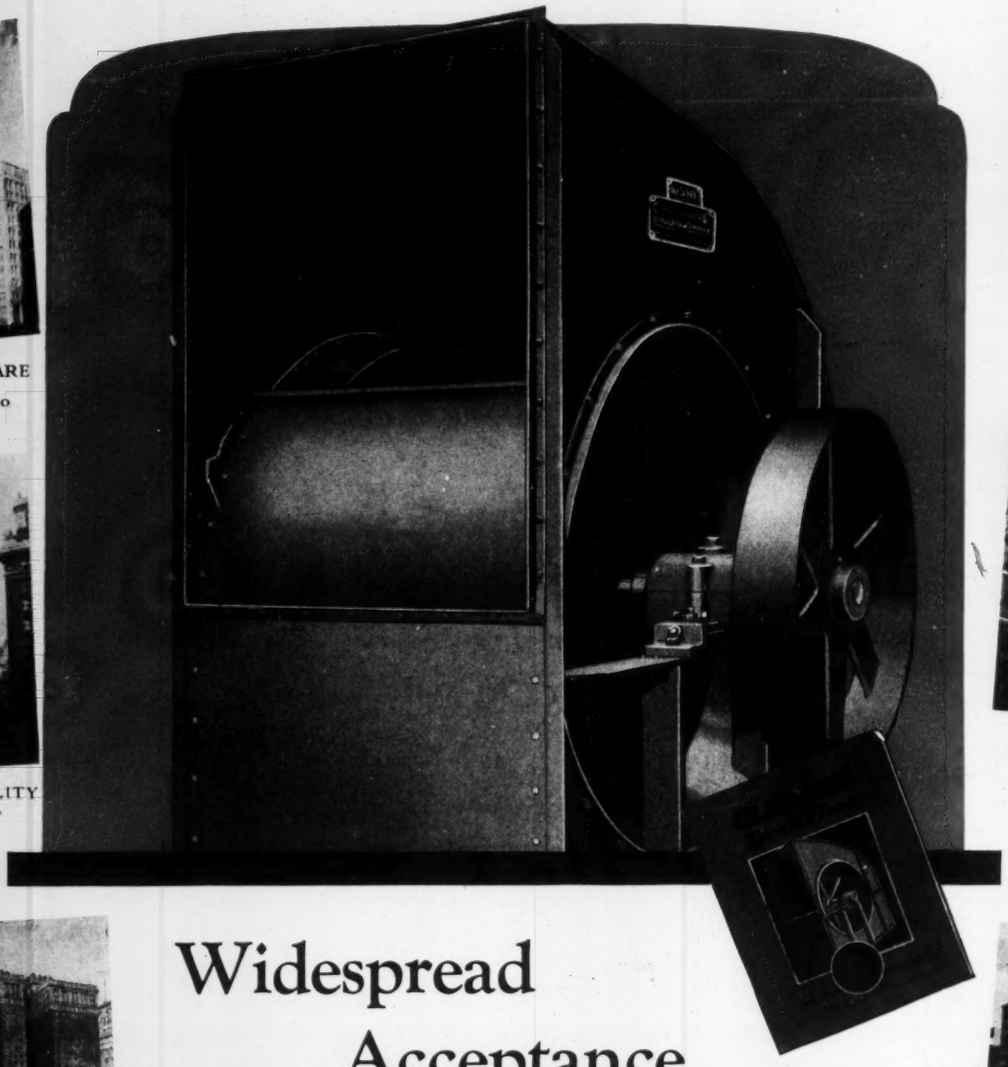
By A. W. THOMPSON, S. R. LEWIS
and MALCOLM C. W. TOMLINSON

Journal Section
of A. S. H. V. E.

October, 1929

FOUNTAIN SQUARE BUILDING, Cincinnati, Ohio

PHILADELPHIA–FIDELITY TRUST BUILDING, Philadelphia, Pa.



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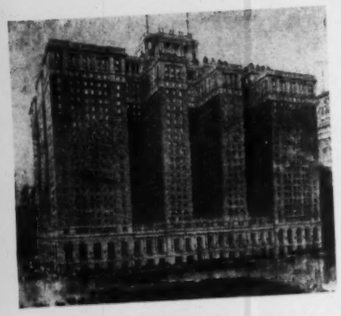

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OLDS HOTEL, Lansing, Mich.

WEST VIRGINIA STATE CAPITOL, Charleston, W. Va.

PALMER HOUSE, Chicago, Ill.

MT. VERNON HIGH SCHOOL, Los Angeles, California.

MITCHELL SCHOOL, Denver, Colo.

The EDITOR'S PAGE

Heating · Piping and Air Conditioning

SOME DOPE ON CHIMNEYS

Mr. Hattis, consulting engineer of Chicago, has worked out three charts which make the problem of computing sizes for tall chimneys a matter of seconds instead of hours.

We sent advance proofs of the article to A. B. Martin of the Kewanee Boiler Corporation asking for his comments. This is Mr. Martin's reply:

"It is very apparent that this paper is the result of considerable work and careful thought on the part of the author. It will provoke discussion, as all such papers do, which may result in Mr. Hattis working out a plan for even a further simplification of material or information which is in almost constant demand and where the simplest possible formula for use by engineers is of incalculable benefit.

"We might add that we have taken the same examples worked out by the author and made calculations on available draft according to other rules. In some cases there has been a very slight variation, but the result so closely approximates that reached through the medium employed by Mr. Hattis that we would not consider it important.

Homer R. Linn, chief engineer, western executive office of the American Radiator Company, also comments on Mr. Hattis' article:

"Data on chimneys can, of course, be obtained by methods other than those shown in this article. For this reason, Mr. Hattis' paper is likely to provoke some lively discussion. It is hoped that it will do so.

"I have gone over the article carefully, and it would appear that the author has given the matter much careful study. I have felt very keenly the need for such a paper because of the many times the heating plant is blamed when the real fault is in the chimney.

"I anticipate this article and the discussions which will follow will render a much needed service to the heating trade."

IN SIX MONTHS HEATING, PIPING AND AIR CONDITIONING HAS WON LEADERSHIP

It would not be unusual, nor yet indicative of a lack of progress, should a publication take six years to reach the position that HEATING, PIPING AND AIR CONDITIONING has won for itself in six months. This issue goes to the heating, piping and air conditioning field with the prestige obtained by its five predecessors. The October number, we feel, will add greatly to the position that HEATING, PIPING AND AIR CONDITIONING already occupies in its field.

Numerous articles which will maintain the high standards established by the technical pages of these first six issues are at present in shape for publication. Many others are in preparation. *Air Conditioning of a Flour Mill,* by Edgar S. Miller; *Air Conditioning and Refrigeration of a Large Bakery,* by W. L. Fleisher; *Controlled Indoor Atmosphere,* by Perry West; *Controlling Weather in a Cigar Factory,* by R. D. Touton of Bayuk Cigars; and *Air Conditioning a Retail Sales Room,* by F. B. Orr of the Commonwealth Edison Company, are a few of the titles scheduled for publication in the near future.

Papers by equally well-qualified writers in the heating and piping fields will be presented for the approval and information of our readers. We feel fortunate that our list of writers includes such names.

ADAMS, THOMPSON AND REDFIELD APPEAR IN THIS ISSUE

We again offer to our readers papers by prominent engineers making their premier in the October number of HEATING, PIPING AND AIR CONDITIONING. N. D. Adams, superintendent of the Franklin heating station at Rochester, Minn., has written an article describing the plant in detail and presenting the reasons for its construction and location. Mr. Adams is preparing a series of papers on the heating and air conditioning of the Mayo clinic, which the Franklin heating station serves, for publication in future issues.

Another new name this month is that of A. W. Thompson, of the Parks-Cramer company. Mr. Thompson's article on computing the capacity of a humidifying plant will prove of value, we are certain, to all engineers connected with air conditioning work. It should also be of interest to plant engineers concerned with humidification.

S. B. Redfield, assistant chief engineer of the Ingersoll-Rand Co., has contributed an article that describes in detail the procedure of computing piping to air compressors and hammers. Mr. Redfield also considers the proper capacity for air receivers. Several time-saving methods of making the computations are discussed in the article also.

A Study of Pipe Joints

The third and concluding article of the series on the design of flanged pipe joints is published in this issue. Arthur McCutchan, associated with Sabin Crocker in the engineering department of the Detroit Edison Co., collaborated in the preparation of the present installment of this valuable work.

The World Congress

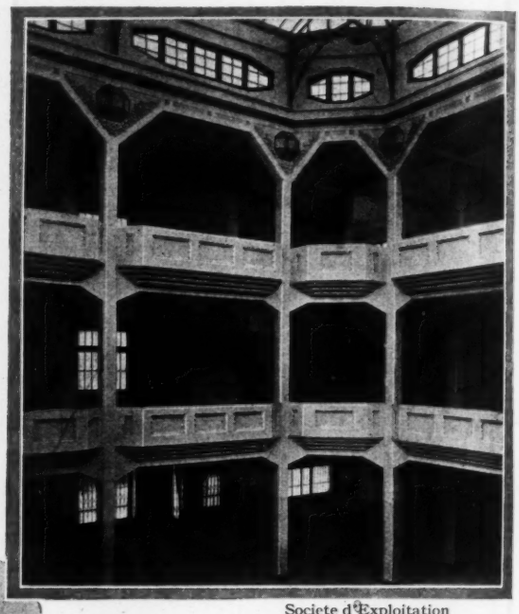
We can't help but envy the engineers sailing this month to attend the World Congress in Japan. It must be a severe mental strain to have to drop business and professional cares in order to run over to Tokio for such a meeting.

The paper to be presented by W. H. Carrier, and prepared by him and a cooperating committee, will be published in complete form in an early issue of HEATING, PIPING AND AIR CONDITIONING. The advisory committee, appointed by the American Society of Heating and Ventilating Engineers, was composed of A. C. Willard, W. L. Fleisher, S. R. Lewis, F. R. Still, O. W. Armspach, H. P. Gant, C. P. Yaglou, H. W. Ellis, F. C. Houghten, and L. A. Harding. A complete list of the papers to be presented at the Congress was published last month.

A news item on the Congress appears in this month's journal.

from Broadway *to the* Takou Road

Paramount Theatre and Building, 42nd and Broadway, New York, "The Cross-Roads of the World"



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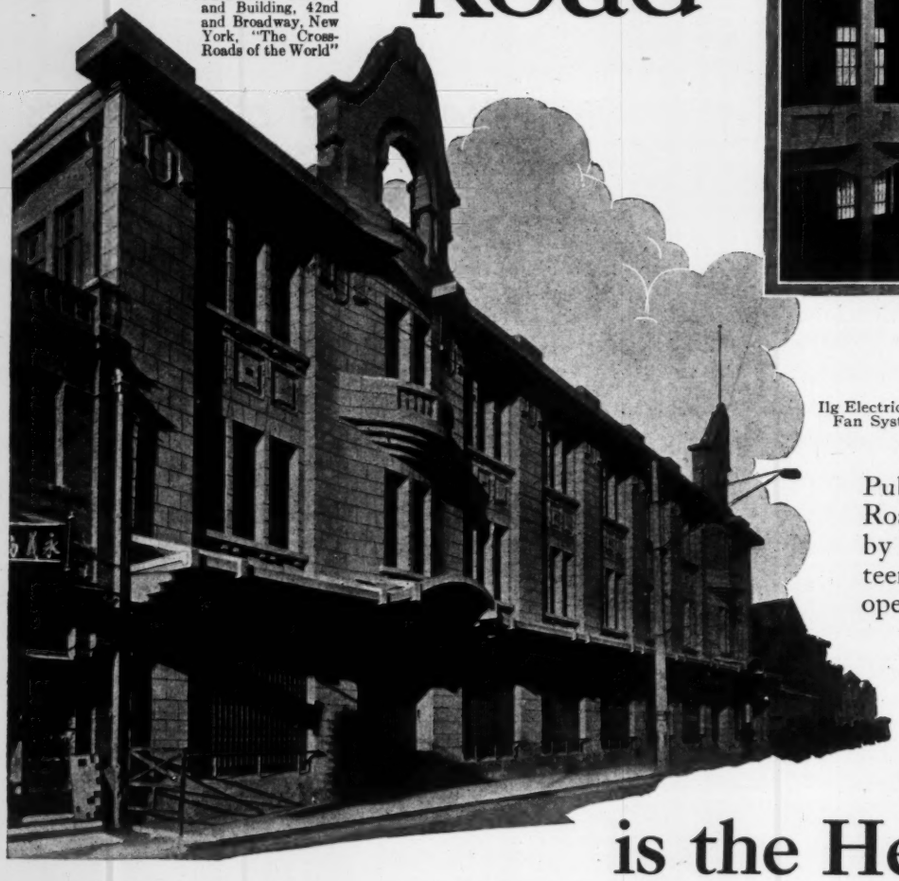

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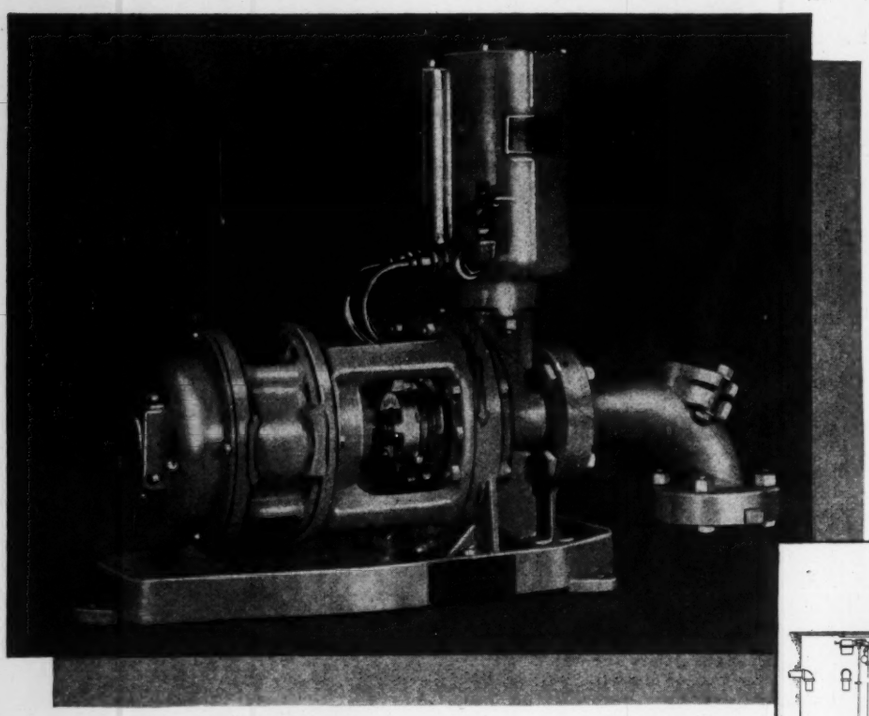

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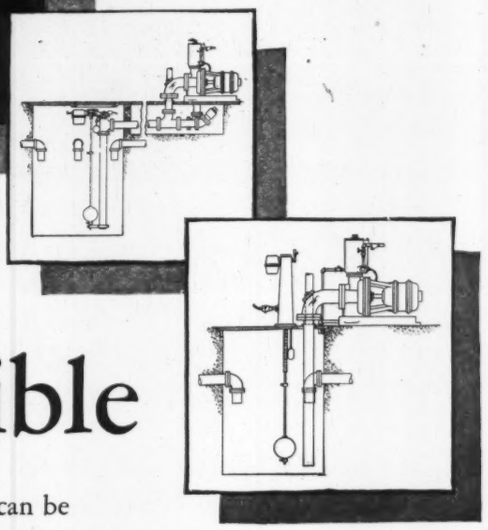

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THERE is just one way that the ordinary sump pump can be installed. The pump must be placed in the pit, the motor directly above. Often this proves awkward and inconvenient.

The Jennings Suction Sump Pump is so designed that only its suction pipe need be submerged. A bronze air rotor, mounted on the same shaft with the water impeller, permits the assembly to handle air or gas, along with seepage, without becoming air bound or losing its prime.

The advantages of this arrangement are obvious. The Jennings Suction Sump and its driving motor are placed outside the pit and can be installed either nearby or removed from it. Space above the pit may be left vacant for other uses. All moving parts are above the floor level where they are always accessible. In addition, the pumping unit is so constructed that the impeller is removable, for inspection or cleaning, without breaking any pipe connections or disturbing shaft alignment.

RETURN LINE AND AIR LINE VACUUM HEATING PUMPS—CONDENSATION PUMPS—COMPRESSORS AND VACUUM PUMPS FOR AIR AND GASES—STANDARD AND SUCTION CENTRIFUGAL PUMPS—HOUSE SERVICE PUMPS—SEWAGE EJECTORS—SUMP PUMPS—FLAT BOX PUMPS—MARINE PUMPS

Jennings Pumps

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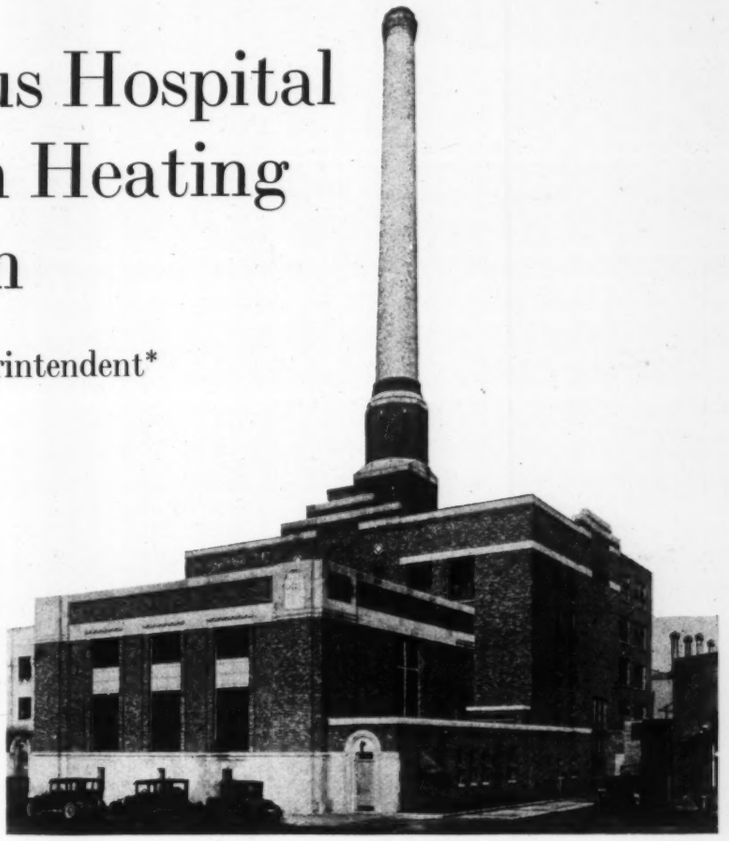
Heating · Piping and Air Conditioning

Vol. I | October, 1929 | No. 6

Serving a Famous Hospital —The Franklin Heating Station

By N. D. Adams, Superintendent*



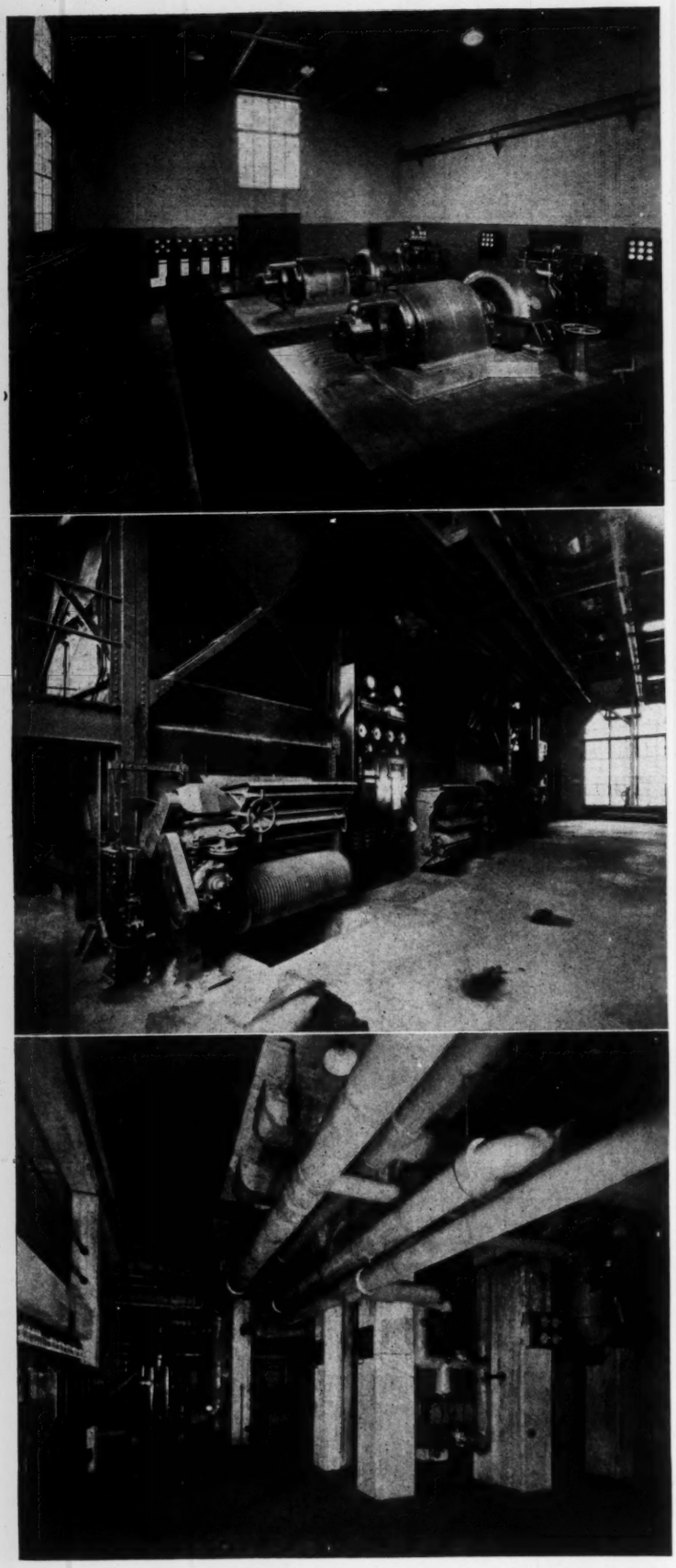
IN 1913 the Mayo Clinic at Rochester, Minn., completed and opened its original building which its owners thought would be large enough for their business for all time. But within two years the building had been filled to capacity and extensions were then made into adjoining properties rented from time to time as the business expanded. Immediately after the world war the Mayo Clinic realized that additional building space must be provided, but wished to wait a few years to study developments following the reconstruction period.

During the winter of 1926 and 1927 the decision was reached to proceed immediately with plans for a new building. Dr. Henry S. Plummer was appointed chairman in charge of the building committee and commissioned Ellerbe and Co. of St. Paul and Rochester as the architects and engineers. This building is now completed and was opened on the first of last May.

The Kahler Corporation operates a group of hotels and hospitals to care for the people coming to the Mayo Clinic for diagnosis and medical treatment. On September 21, 1921, they

Mr. Adams has agreed to describe in a later issue the heating, piping and air conditioning of the new Mayo Clinic in Rochester, Minn., which is a fine example of the latest in engineering in this field. The combined water, steam and power plant, known as the Franklin Heating Station, serves this building as well as other buildings at this famous Rochester institution. A knowledge of its design and operation is essential to a complete understanding of the systems in the Clinic itself, and this article is presented first for that reason.

* Formerly mechanical engineer, Ellerbe & Co., St. Paul. Member A. S. H. V. E. and Engineers' Society of St. Paul.



opened The Kahler, the largest and most beautiful of these.

The original Mayo Clinic building, as well as each building owned by the Kahler Corporation, contained a high pressure boiler heating plant for its own needs. The heating plant in the new Kahler Hotel had been extended to heat the Curie Hospital, which is adjacent, and to furnish steam at 90 lb. for sterilizing purposes to the Clinic building and to the Colonial Hospital during the summer.

With the decision to build the new Clinic building, it was necessary to decide whether or not this building would contain a heating plant. The heating plant in the existing building was not large enough to take care of additional service and the room available was not sufficient for additional equipment. After the engineers with the architects had made several studies of the proposed new building they reported to the owners that the diversified services required by the properties would make it advisable to install a high pressure boiler plant, with turbo-generators for supplying the electrical demand.

The engineers were advised to study and report on the advisability of a combined central heating plant for the Clinic and Kahler Corporation properties.

This report was submitted during February of 1927, and the engineers were instructed to proceed with the plans for the present Franklin Heating Station.

Location of Station

The location of the Franklin Heating Station was chosen after considering two sites:

(1) Near the Great Western railroad with siding.

(2) Present location. The property being owned by the Kahler Corporation.

The points for and against the two sites were:

(1) Disadvantages:

a—Problem of obtaining option on site desired.

b—Possibility of not being able to obtain easements for service tunnel between plant and properties.

c—Should a property owner along the right of way demand service in exchange for easement, the plant would then come under the regulations governing public utilities.

TOP—INTERIOR VIEW OF TURBINE ROOM, SHOWING CLEARLY THE BLEEDER CONTROL MECHANISMS WITH LINKAGE TO MAIN GOVERNOR, THE MAIN ELECTRICAL CONTROL PANELS AND THE RECORDING AND INDICATING PANELS. MIDDLE—INTERIOR BOILER ROOM FIRING AISLE, SHOWING STOKERS, INDICATING AND RECORDING PANELS, WEIGH LARRY AND BOTTOM OF COAL AND ASH BUNKERS. BOTTOM—BASEMENT VIEW, SHOWING FOUNDATIONS, CONDENSERS, AIR COOLER FOR GENERATOR, DEEP WELL PUMP WATER BOOSTER PUMPS, LINES LEADING INTO SERVICE TUNNEL AND AUTOMATIC SWITCHING EQUIPMENT

Advantages:
a—Coal handled directly from cars to overhead bunkers in plant.
b—Lower first cost of housing and desirability of eliminating chimneys from the business sections.

(2) Disadvantages:

a—Cost of coal handling by truck.
b—Additional cost of housing and equipment. The architecture of the building would necessarily be more expensive to harmonize with adjacent buildings.
The equipment would have to be designed to operate as quietly as possible due to adjacent hotels and hospitals.
c—Smoke problem.
Advantages:
a—Owners' property.
b—Continuity of properties separated only by streets and alleys.
c—Reduced length of service tunnels.
d—Reduced transmission costs.
e—Closer contact between supply and demand.
f—Possibility of working out a combined water works, steam and power plant.

A study of the advantages and disadvantages of each location clearly indicated that the site chosen, although higher in first cost, would give the most economical over-all operation. The deciding factors were:

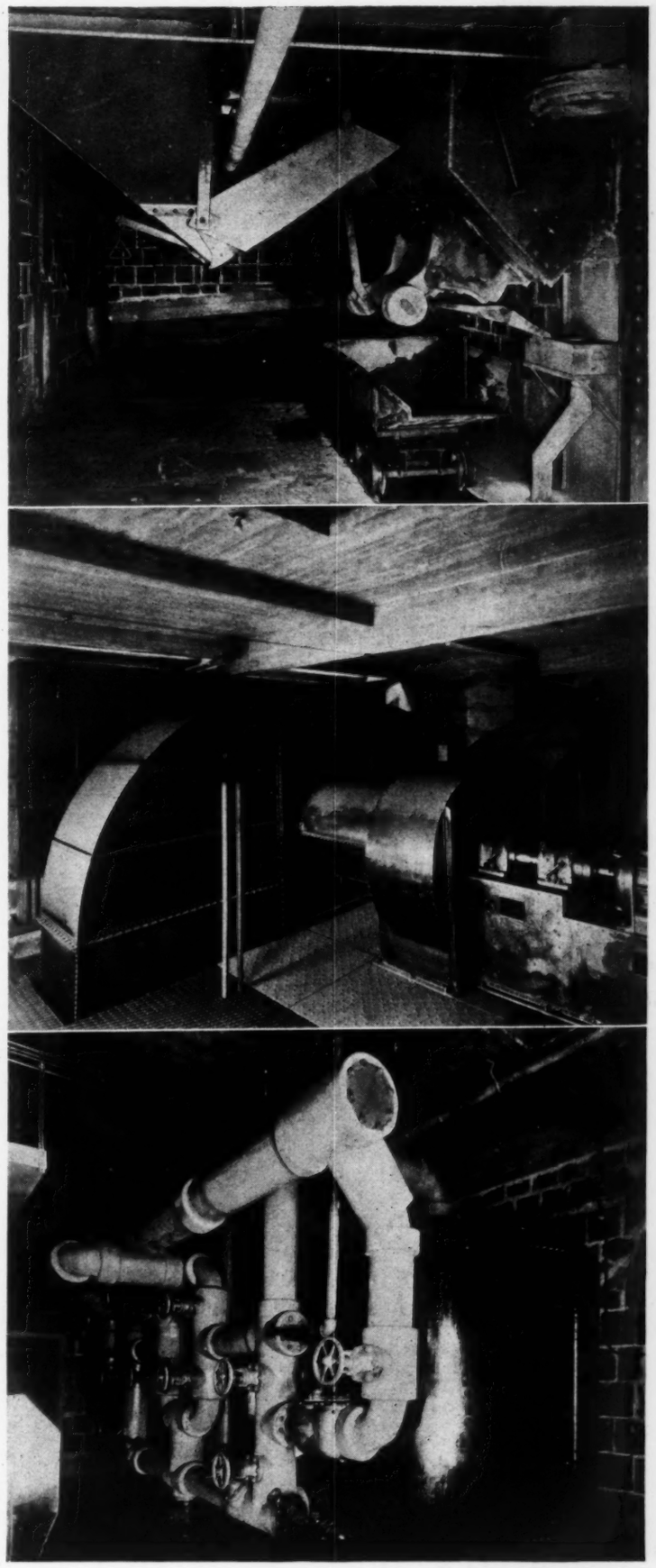
a—Freedom from outside interference.
b—Closer contact.
c—Hauling from cars or silos, with a large storage yard for winter emergency, is more economical than transmission losses from the railroad site.
d—Heat balance obtainable by combined water, heating and electrical systems.

Coal Handling

The coal is bought directly from the mines in Illinois. The plant is now operating on Saline County coal ¾-in. screenings, which run very uniform in grading and quality.

Average Analysis

	Per cent as in laboratory	Per cent dry basis
Moisture	7.44	
Ash	11.66	12.60
Volatile	31.70	34.25
Fixed carbon	49.20	53.15
	100.00	100.00
B.t.u.	118.44	127.96
Sulphur	2.66	2.87



TOP—ASH TUNNEL SHOWING ASH AND SIFTINGS HOPPERS, TRACK AND ASH CAR. MIDDLE—VIEW OF INDUCED DRAFT FAN, AIR PREHEATER, BREECHINGS AND BYPASSES. BOTTOM—WATER SOFTENING PLANT FOR ALL HOT WATER TO BUILDINGS

LONGITUDINAL SECTION THRU FRANKLIN HEATING STATION - ROCHESTER, MINN.

THE FRANKLIN HEATING STATION SERVES THE MAYO CLINIC AT ROCHESTER

Six to ten (6 ton) truck loads are hauled to the plant each day between 7 and 10 A. M. The coal is dumped into a receiving hopper from which a skip hoist automatically receives, raises and discharges into a larry which distributes lengthwise of the overhead bunker.

The bunker is a cast iron sectional of 400 ton capacity. From the bunker the coal is handled through a traveling larry to the stoker hoppers.

Directly beneath the boilers the 14-ft. wide ash tunnel runs the full length of the boiler room. Each boiler is equipped with an ash and siftings hopper with balanced ash gates of the guillotine design. An industrial track is installed under the hoppers, running lengthwise of the tunnel and terminating at the skip hoist. The ashes are discharged into an ash car which is pushed to the skip hoist and emptied into the skip bucket, which in turn automatically rises, dumping into the ash bunker which is located at the north end of the coal bunker.

The gate for removing the ashes from the bunker is directly above the coal receiving hopper. After a truck has delivered a load of coal it is filled with ashes for which there is a continuous demand for grading and filling. This pays for their removal.

Boiler Equipment

Boiler equipment consists of three 400 h.p. water tube boilers equipped with partial water wall furnaces. The boilers are designed and have been operated at 200 per cent continuously, with 250 per cent peaks. On July 17 last the stoker chain on boiler No. 2 broke at about 7:30 A. M., just before the morning load came on. Before boiler No. 3 could be brought on to the line, boiler No. 1 was forced to carry a little over 300 per cent its rating.

A chain grate stoker 6 x 13 ft. is installed under each boiler. The furnaces are designed with arches and, over the fire, air inlets in order that smoke may be eliminated. Due to the plant's being located in the central area it is absolutely necessary that all smoke be eliminated. Therefore, this was the major thought in the entire furnace design and has worked out exceptionally satisfactorily. Each boiler is equipped with a tube type air preheater which raises the temperature of the air entering the furnace to approximately 300 deg. fahr. The gases leaving the air preheater and entering the chimney are reduced approximately 125 deg. below entering temperature.

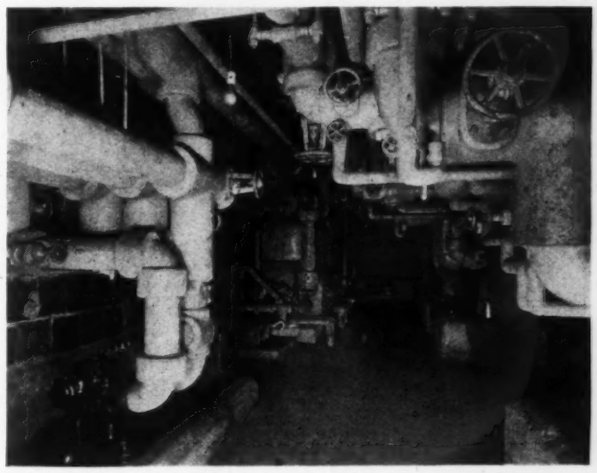
Forced draft fans draw the air through the preheater and force it into a horizontal main air duct that runs lengthwise of the boiler room so that there may be flexibility in use of the fans on each boiler. The forced draft fans are equipped with steam turbines. The induced draft fans are located on the fifth floor level and are equipped with slip ring motors for speed control. One boiler feed pump located in the central piping gallery in the basement is equipped with an induction motor and the other with a turbine. This arrangement of turbines and motors gives flexible control, continuity of service and a steam balance in the summer time.

During the winter time all of the exhaust steam can be utilized for heating with all of the turbine equipped machines in use, but during the months of July and August it is necessary to operate motor equipped machinery and use only a sufficient number of turbines to give the steam required for the boiler feed water heating.

Turbines

The two turbo-generators were specially designed for this plant. They are of the bleeder type with extractions at the 90 and 10 lb. pressure stages. The operation is as follows:

The steam is supplied to the governor valve at 250 lb. pressure. The first extraction stage is at 75 to 90 lb. with a governor for regulating the steam required for sterilizing, cooking and other process work of variable demands. The 10 lb. extraction is automatically controlled and may be adjusted between 5 and 10 lb. pressure. This steam is sent out through the building to be further reduced to the pressure required for the heating system in each of the individual buildings. The balance of steam required for generating flows into the condenser where a vacuum of 20 to 23-in. is maintained. The water enters the condenser at 48 to 50 deg. fahr. and is automatically controlled at the outlet. The final temperature is not allowed to rise above 130 deg. fahr., as higher temperature affects the vacuum too much for steady action of the throttle governor.



Main Header and Pipe Gallery. View Shows Boiler Feed Pump, Evaporators and Forced Draft Fans

Electricity

Current is generated at 2300 volts, 3 phase, 60 cycle. The generators are equipped with direct connected excitors but an auxiliary turbine driven excitor is installed for emergencies. Each generator is equipped with a closed water cooling system which tends for efficient and quiet operation as well as to keep the windings clean.

Directly below the oil switch gallery a 2300-440 volt sub-station is equipped with transformers, with distributing panels for the station equipment and automatic starting switches with compensation. Overload and under-voltage relays control the circuit to each motor.

A start and stop button is located at each motor

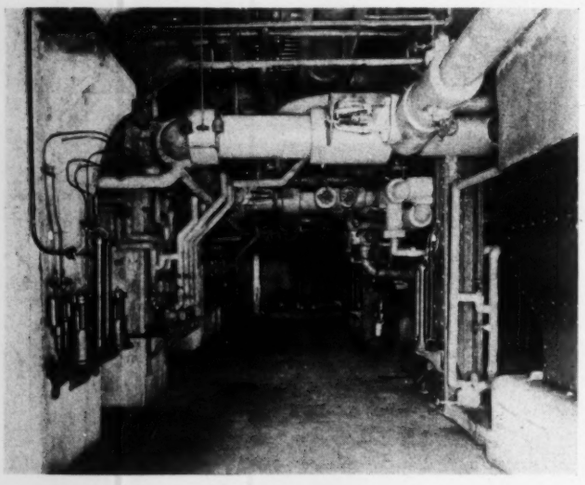
with a lockout switch for protection of workmen while overhauling the machine. Wherever practical, starting circuits are automatically controlled by flood and pressure switches.

The distribution to the buildings is at 2300 volts to A and B sub-stations centrally located to furnish energy to the buildings at 220 volts, 3 phase for power, and 110-220 volts, single phase for light and fractional motor operation. For continuity of service, which is mandatory in hospitals, two feeder cables were installed with automatic switches designed to throw all of the load on to one in case of a failure in the other. This installation was one of the first to be manufactured and was designed by Mr. Frisbie, electrical engineer with Ellerbe & Co., architects and engineers.

Water

Water is obtained from two deep wells located directly below the heating plant. These two wells are 500 ft. deep and 12 in. in diameter up to 150 ft. below the floor, where they increase to 16 in. O. D. well casing to the pump foundation. Two deep well pumps lift the water from the wells and discharge it into a 25,000 gallon storage tank in the upper part of the boiler room. From the storage tank the water flows by gravity down into a header located in the ceiling of the pump room.

Cold water supply for the buildings flows from the main header into the suction of the booster pumps, which force it through the tunnel main with branches to each building. The branch to each building is provided with a strainer, reducing valve and meter. The reducing valve is furnished in order to maintain a constant pressure at the fixtures on the top floor of the building. As an illustration, in order to have sufficient water on the operating floor at the Kahler Hotel it is necessary to maintain 70 lb. pressure at



Detail of Piping Between Header and Turbines. In the Background May Be Seen the De-Superheaters, Hot Well and Condensate Booster Pumps

the reducing valve, while in the Colonial Hospital 40 and 45 lb. is sufficient. The temperature of the water as it comes from the deep wells is from 48 to 50 deg. fahr. and reaches the buildings about 55 or 60 deg. fahr.

For heating and softening, water is taken from the main header in the basement and flows by gravity through the condenser underneath the turbine where it is raised to a temperature of 100 to 130 deg. fahr. From the condenser the water flows through the softeners and then back to the booster pumps and is

View of Coal Gallery Showing Larry for Distributing Coal Along the Length of the Bunker

delivered through the main tunnel to the building. The equipment in each building is the same as described under cold water, except a hot water heater of the closed type is provided to boost the temperature of the incoming water to any temperature desired at the fixtures. This temperature now ranges from 140 to 180 deg.

As the amount of steam to the condenser varies with the electrical load, the 10 lb. heating steam load, and the 90 lb. sterilizing and process steam demand, the temperature of the water leaving the condenser varies between 80 and 130 deg. fahr. The hot water demand from the building varies from 25 or 30 gallons a minute at night to a maximum of 300 or 400 gallons a minute during the day time. The quantity of water flowing through the condenser is determined by the amount of steam to be condensed. When the amount of water going through the condenser is more than that demanded by the buildings, it is discharged to the storm sewer through an automatic temperature relief valve.

Distribution

All service to the various buildings is installed in underground concrete tunnels.

(1) The 2300 volt electrical cables are in fiber conduit which was laid in the wall or floor construction with a sufficient number of pull boxes for the pulling in the splicing of conductors.

(2) Steam for sterilizing, cooking and laboratory work is piped to the buildings at 75 to 90 lb. pres-

sure where it is reduced to the pressure required at the point utilized. This not only takes care of the losses by radiation and line friction but allows flexibility in the purchase and use of different makes of equipment.

(3) Steam for heating is furnished to the buildings at 5 to 10 lb. pressure, according to the demand created by Minnesota's changeable weather. As the heating systems vary in design, one vacuum, another vapor and so on, it was necessary to reduce the steam pressure at each building to suit the conditions existing.

(4) The condensate is collected by vacuum receiving tanks from the heating systems and by flush tanks from intermediate pressure returns and returned by float operated pumps through meters into a common pipe line which discharges into the hot well which is located at the heating plant.

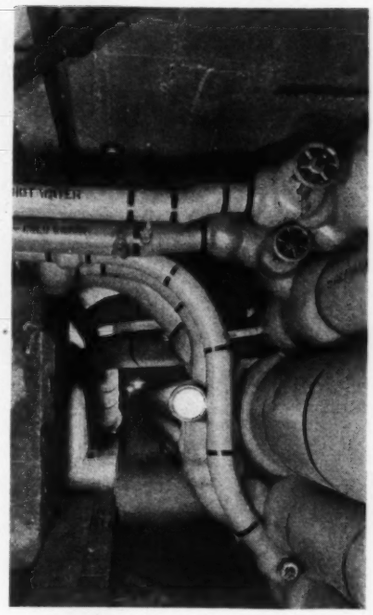


FIG. 11. SERVICE TUNNEL VIEW

(5) Water distribution was covered. All pipe in the distribution system is supported on side wall hangers designed for saving of space, strength and adjustment. Each is equipped with an expansion roller.

All piping is acetylene welded except at branch connections, valves and expansion bends. Bends were used to take care of expansion throughout except in one straight run of 350 ft. where double slip joints were installed. All steam and water lines throughout the plant and tunnels were efficiently covered with magnesia and wool-felt to prevent losses by radiation.

Boiler Feed Water

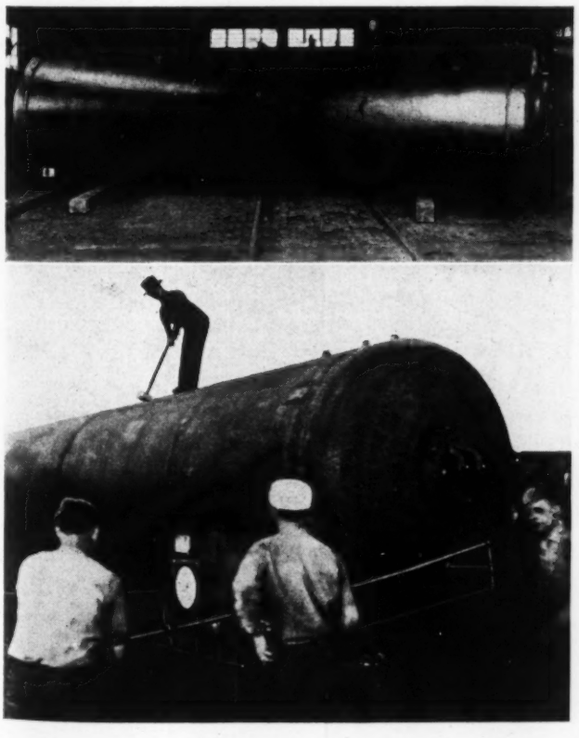
Evaporators are installed to supply the necessary make-up water required for boiler feeding. This amounts to from 10 to 30 per cent and is governed by the amount of return condensate from the buildings.

Design for Welded Pressure Vessels

ENGINEERING investigations made in connection with the development of welded pressure vessels have contributed greatly to knowledge of correct design for such vessels. Previously, mechanical-type joints could not be satisfactorily tested at hydrostatic pressures exceeding about 1½ times the working pressure, because of the serious leakage at the joints. As a result, it was impossible to determine experimentally the exact behavior of vessels at high pressures and designs had to be based on more or less theoretical consideration.

The welded joint, however, imposes no such limitation on pressure testing. Hydrostatic pressure can be carried on up to the point of failure. Strain gage measurements made while the tank is under pressure indicate points of excessive stress and thus show exactly where the weak spots in the design are. Such measurements made on a large vessel fabricated according to previously accepted principles of design showed quite conclusively that there were serious errors in these designs at many points.

Zones of unexpected weakness were found in several important points, notably at the knuckle of the dished heads, at the manhole opening, and at nozzles. It is interesting to note that welding not only disclosed these weak points in design but also provided the means of correcting them. Designs were modified in accordance with the indications of the tests, and the new designs were then submitted to strain gage measurements while under hydrostatic pressure. As a result, it is possible today to build a pressure vessel of known uniform strength at every joint. Under pressure, such a vessel acts as a single unit throughout; there are no concentrations of stress.



ABOVE—6 x 24 TANK WITH SINGLE COURSE SHELL FOR 300 LB. WORKING PRESSURE. BELOW—TESTING BY HAMMERING WELDS WHILE UNDER 400 LB. PRESSURE

Computations for Piping to Air Compressors and Hammers

By S. B. Redfield

AS PRESENTED, the problem is not altogether clear; but to use it as a basis of an explanation of the method of figuring, certain assumptions will be made. Changes in these asumptions would change the answer, but the principles would remain the same.

In the first place, the "capacity" of the hammers is not definite. This could be displacement per stroke, or it could be an attempt to express the amount of free air that would be used per stroke and to get this, the pressure existing in the hammer cylinder at the time the exhaust opens must be assumed. The next question is, does "per stroke" mean per down-stroke, or per each stroke? We would interpret it to include one complete cycle from the beginning of one down-stroke to the beginning of the next. It would then include one up-stroke also.

Not knowing the dimensions of the hammer, we will arbitrarily assume a set of conditions which could give about the capacity stated, thus illustrating how such a figure may be arrived at by calculation. All figures given are by slide rule calculation, to nearest third place figure only.

PROBLEM

"I would like a little information. I want to install an air receiver to have sufficient air for running two hammers in our forge shop. One is a 1500 lb. hand hammer with a 2 cu. ft. capacity and 100 strokes a minute. The other a forging hammer 2½ cu. ft. air capacity and 85 strokes a minute and will idle 50 per cent of the running time.

"We have two air compressors, one delivery 145 cu. ft., the other delivery 609 cu. ft. free air per minute at 100 lb. pressure. The hammers are about 80 feet from the compressors; hammer air inlets 2 in. pipe, 15 ft. long with globe valves, 2 in. pipe reduced from 4 in. pipe.

"What will be the pressure drop, and how long could the air be retained in the air receivers before the pressure drop is so great that the hammers could not be used effectively?

"How are these computations made?"

Assume hammer piston diameter 8 in., with a 3 in. piston rod on lower side only. Assume a 10 in. average stroke. After the air has been admitted to the hammer cylinder, it pushes the piston down and, due to the valve action, there is some expansion, so that when the exhaust opens, the pressure of the air in the hammer cylinder is something less than the admission pressure. For our example, let us assume the release pressure is 45 lb. per square inch on the down-stroke and 25 lb. on the up-stroke.

First, to get the displacement of the upper side of the hammer piston, per down-stroke:

Area 8 in. diameter	=	50.2
Stroke		10
Down displacement		502 cu. in.

To get displacement of lower side of hammer piston:

Area 8 in. diameter	=	50.2
Area 3 in. rod (Approx.)		7.1
		43.1
Stroke		10
Up displacement		431 cu. in.

At end of down-stroke, at time of opening of exhaust, the amount of air in the space above the piston is, of course, the displacement, or 502 cu. in.; but this air is still compressed to 45 lb. gage pressure by our assumption. In pneumatic calculations, all air quantities are expressed as "free air," or air at atmospheric pressure and temperature, because that is the kind of air which the compressor must take in.

For our problem, which can not be accurate because we are, and always would be, forced to make numerous assumptions, we can neglect temperature changes. If the 502 cu. in. of top displacement of the hammer piston were filled with air at end of stroke at 45 lb. pressure, this would be equivalent to

$$\frac{502}{1728} \times \frac{(45 + 14.7)}{14.7} = \frac{502 \times 59.7}{1728 \times 14.7} = 1.18$$

cubic feet if expanded to atmospheric pressure. This is because, neglecting temperature changes, the volume of compressed air would increase in inverse proportion to the absolute pressure. The absolute pressure at release was assumed to be;

$$45 + 14.7 = 59.7$$

and after expansion to atmosphere, its absolute pressure would be 14.7 lb. per sq. in. The volume would increase, therefore, in proportion to $\frac{59.7}{14.7}$ when expanded to atmospheric pressure.

In the same way, the air on the under side of the

piston, which we have assumed to have a pressure of 25 lb. at the release point, would amount to

$$\frac{431}{1728} \times \frac{(25 + 14.7)}{14.7} = \frac{431}{1728} \times \frac{39.7}{14.7} = .674 \text{ cu. ft.}$$

when expanded to atmospheric pressure. Thus one down-stroke and one up-stroke, or a cycle, would consume

$$1.18 + .674 = 1.85 \text{ cu. ft.}$$

of "free air." We have made no allowance for air wasted in hammer cylinder clearance nor in leakage, so to be on the safe side, this would certainly be assumed to be 2.00 cu. ft., as in the original problem.

In the same way, the free air consumption of the other hammer per cycle could be worked out, and we will assume the figure given, 2½ cu. ft., means the approximate amount of free air used for one down-stroke and for one up-stroke. The problem states that this larger hammer "will idle" 50 per cent of the time. In steam practice, this is literally true; that is, the hammer probably would be left to run up and down continuously of its own accord, to keep the pipe line clear of condensation. It is just in this point that an air hammer is economical; an air hammer would not run idle. It would be shut down completely when idle. The problem also states it will idle 50 per cent of the running time. This could mean it would run two minutes and then be idle one minute, or half of the length of time it had previously run; or we could assume it means it runs one minute and is idle one minute, or 50 per cent of the total time. We will take the latter interpretation.

In any plant the time factor is of great importance. When there are numerous machines to be operated, an average condition may be adopted which will cover the requirements quite well, the average being much less than the sum total of all the machines operating at once. However, when there is but one machine, or two, or even three, and they may all operate at one time, the air consumption during that time will be the sum total and the plant air capacity must be equal to that sum total. In this problem, with both hammers likely to operate at the same time, the idle time of one machine would not be considered when figuring compressor requirements. When one or both machines are idle, the unloading mechanism on the compressor will reduce the compressor output and the horsepower to drive it, so that the idle time of the hammer is a real economy in the power bill, but the full capacity is available when both hammers are running together. For the case in the problem, the total capacity would be, from the assumptions made, and for 100 strokes for the one and 85 strokes for the other hammer:

$$2 \times 100 = 200$$
$$2\tfrac{1}{2} \times 85 = 213$$

Total Free Air 413 cu. ft. per min.

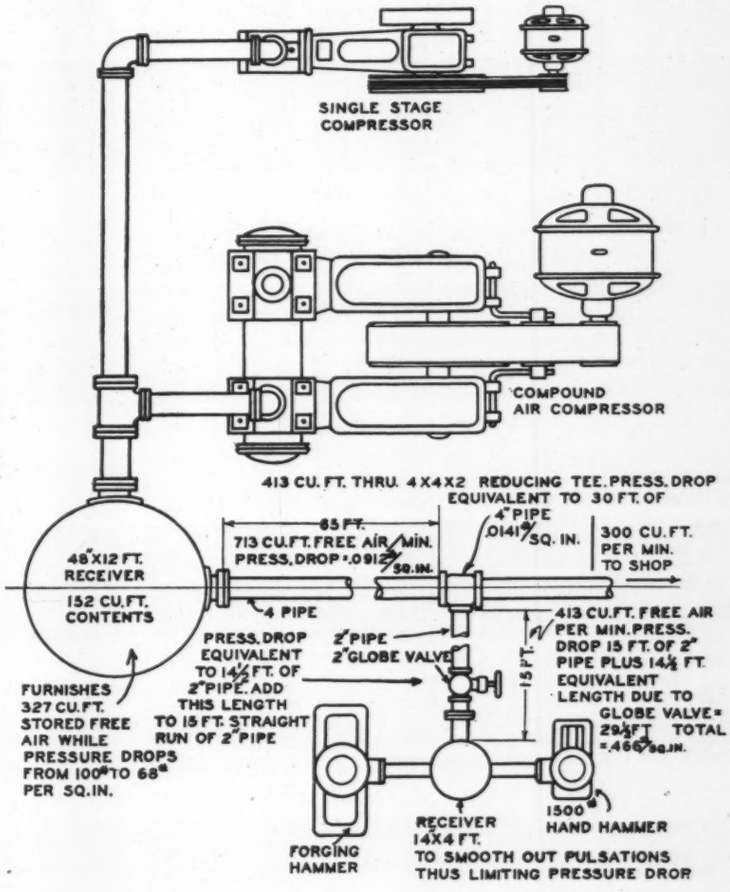

SINGLE STAGE COMPRESSOR
COMPOUND AIR COMPRESSOR
413 CU. FT. THRU. 4X4X2 REDUCING TEE. PRESS. DROP EQUIVALENT TO 30 FT. OF
4" PIPE .0141 SQ. IN.
65 FT.
713 CU. FT. FREE AIR/MIN. PRESS. DROP .0912 SQ. IN.
300 CU. FT. PER MIN. TO SHOP
48"X12 FT. RECEIVER
152 CU. FT. CONTENTS
4 PIPE
413 CU. FT. FREE AIR PER MIN. PRESS. DROP 15 FT. OF 2" PIPE PLUS 14½ FT. EQUIVALENT LENGTH DUE TO GLOBE VALVE = 29½ FT. TOTAL = .466 SQ. IN.
PRESS. DROP EQUIVALENT TO 14½ FT. OF 2" PIPE. ADD THIS LENGTH TO 15 FT. STRAIGHT RUN OF 2" PIPE
2" PIPE
2" GLOBE VALVE
15 FT.
FURNISHES 327 CU. FT. STORED FREE AIR WHILE PRESSURE DROPS FROM 100# TO 68# PER SQ. IN.
1500# HAND HAMMER
RECEIVER 14X4 FT.
FORGING HAMMER
TO SMOOTH OUT PULSATIONS THUS LIMITING PRESSURE DROP

Diagram of Piping Showing Pressure Drop

Compressor Output

In the problem it is stated that one compressor has a delivery of 145 cu. ft. and the other has 609 cu. ft. per minute. Although it states "delivery,"

the probabilities are that these figures really are piston displacement, which would be figured exactly like the piston displacement of the hammer was figured. The actual delivery of the compressor, expressed in cu. ft. of free air per minute would be less than the displacement and would be found by multiplying by the volumetric efficiency.

Undoubtedly the smaller compressor pumps its air to 100 lb. pressure in a single stage of compression; that is, it has but one cylinder. It is also probable that the 609-ft. machine is of the two-stage, or compound type; that is, it employs two compressing cylinders, one large one and one smaller. The larger one takes in atmospheric air and delivers it to an intercooler at a pressure somewhere between 25- and 30-lb. gage. In the intercooler the heat of compression is removed and the air goes to the second-stage cylinder which boosts its pressure to the 100 lb. discharge line. By this two-stage process and the cooling between stages, the power required to compress a given amount of air to a given pressure is reduced, thus improving the economy. The effective piston displacement of the compound unit is of course only that of the first stage cylinder.

The volumetric efficiency of a compressor depends upon several things, the most important being the cylinder clearance and the pressure. The effect of cylinder clearance is to trap some of the air in the waste spaces, instead of discharging it. The re-expansion of this trapped air at beginning of suction stroke does useful work to help the driving motor so is *not a power loss;* but the amount of air delivered is always less than the piston displacement of the cylinder. The power required by the compressor depends upon the amount of air actually delivered and not upon the piston displacement in itself.

For our case, not having actual data, we can assume the volumetric efficiency of the 145-ft. single-stage compression is about 64 per cent. For the 609-ft. compound unit we can assume, say 84 per cent.

From the problem and the volumetric efficiencies assumed, we now have the probable actual delivery of air to the receiver:

$$145 \times .64 = 93$$
$$609 \times .84 = 511$$

Total free air to receiver 604 cu. ft.

per minute. As the two hammers running at the same time will use only about 413 cu. ft. of free air per minute, the compressor capacity exceeds the consumption, so, if no other air is used, there is no problem of receiver storage. This side of the problem will be considered later in this article, however, to show how it would be calculated.

Pressure Drop in Pipe

The next point is the pressure drop in the pipe line. There are numerous formulas for this, but the one usually given in handbooks, for small pressure drop, is

$$p = .0761 \frac{LQ^2_1}{c^2 d^5 r}$$

where

p = pressure drop, pounds per square inch.
L = length of pipe in feet.
Q_1 = cu. ft. of *free air* per minute flowing.
d = pipe diameter, inches.
r = ratio, absolute pressure in pipe line divided by absolute atmospheric pressure.
c = experimental coefficient. This is usually taken from the well known D'Arcy experiments on flow of water in pipes. These values are as follows:

VALUES OF c BY D'ARCY

Pipe diameter..	1-in.	2-in.	3-in.	4-in.	6-in.	8-in.	10-in.
Values of c....	45.3	52.7	56.1	57.8	59.5	60.7	61.8

In the problem, there seems to be a total pipe length of 80 ft., consisting of 65 ft. of 4-in. pipe and 15 ft. of 2-in. pipe. Assuming all the air to both hammers goes through both pipes in series, and realizing that a 4-in. standard pipe is 4.03 in. inside diameter, the pressure drop through the 4-in. pipe figures:

$$p = .0761 \frac{65 \times 413^2}{57.8^2 \times 4.03^5 \times 7.8} = .0306$$

lb. per sq. in., which is negligible.

Similarly, the pressure drop produced by 413 cu. ft. of free air through 15 ft. of 2-in. pipe would be (2-in. pipe being 2.07 I.D.)

$$p = .0761 \times \frac{15 \times 413^2}{52.7^2 \times 2.07^5 \times 7.8} = .237$$

or about ¼ lb. per sq. in.

This is an excellent example of what pipe diameter means in pressure drop. Here we find that 65 ft. of 4-in. pipe produces a pressure drop for 413 cu. ft. of free air per minute, of only .03 lb. per sq. in.; whereas 15 ft. of 2-in. pipe produces a pressure drop of about ¼ lb. The reason is, as shown by the formula, the pressure drop varies inversely as the fifth power of the pipe diameter. The fifth power of 4 is a little over 1,000; whereas the fifth power of 2 is only 32, and 1,000 is more than 30 times 32. Thus it is clear that for equal lengths and same air quantity the 2-in. pipe would produce approximately 30 times the pressure drop that a 4-in. pipe would give. In this case the 2-in. pipe is only about ¼ the length of the 4-in. pipe, so the drop would be only about ¼ as much as for the full length. Instead of the drop being 30 times as much in this problem, therefore, it would be nearer ¼ of 30 or 7.5 times as much for the 2-in. pipe as for the 4-in. Thus, the 2-in. pipe 15 ft. long should give about 7½ times the drop of the 65-ft. length of 4-in. pipe. This would be

$$.03 \times 7.5 = .225 \text{ lb. per sq. in.,}$$

which is very nearly ¼ lb. The actual difference in the figures is, of course, due to using two different coefficients for the two sizes.

This discussion is given to point out how approximations may be made from one set of conditions to another very easily and quickly, by noting how the various factors in a formula affect the result.

Effect of Fittings

Another factor entering into pipe line pressure drop is the effect of elbows, valves, etc. Many handbooks give values for equivalent lengths of pipe which give the same pressure drop that various fittings do. In *Mark's Mechanical Engineers' Handbook,* it states that a 90 deg. screwed elbow gives a loss equivalent to a straight pipe 30 diameters long; turning at a tee is equivalent to 60 diameters of straight length and a globe valve gives about 85 diameters; an angle valve about 40 diameters. A gate valve gives only about 11 diameters, so it is clear that a gate valve is to be preferred to a globe valve in the ratio of about 85 to 11, or almost eight times, as far as line resistance alone is concerned.

In our problem, there doubtless is a tee connection between the 4-in. and the 2-in. pipes. If the side outlet were also 4-in., we have said its pressure drop would be equivalent to 60 diameters or 240 in. of 4-in. pipe, which is 20 ft. As the side outlet is reduced to 2 in., we can well afford to add 50 per cent to this and say the tee is equivalent to 30 ft. of the 4-in. pipe. Then instead of figuring on 65 ft. of 4-in. pipe, we should say 95 ft. Using the same formula as before for the 4-in. pipe; but using 95 ft. instead of 65, the pressure drop would of course be $\frac{95}{65}$ of our first figure. This would then be

$$\frac{95}{65} \times .0306 = .0447 \text{ lb. per sq. in.}$$

It still is negligible.

Similarly, with a globe valve in the 2-in. line, the equivalent additional length of pipe would be 85 diameters, or

$$\frac{85 \times 2}{12} = 14.2 \text{ ft.; say } 14\tfrac{1}{2} \text{ ft.}$$

Of course this must be added to the 15 ft. of straight 2-in. pipe, making a total of say 29½ ft. The pressure drop will then be $\frac{29.5}{15}$; nearly double its former value or

$$\frac{29.5}{15} \times .237 = .466 \text{ lb. per sq. in.}$$

This is nearly ½ lb. and, offhand, this indicates that it would be worth while to increase this 2-in. pipe to 2½ in. Such an increase, as we showed before, would reduce the pressure drop about as the 5th power of the diameter, or as

$$\frac{2^5}{2.5^5}$$

which is about as $\frac{32}{98}$, or roughly 1/3. Thus with 2½-in. pipe the drop would be about .15 lb. per sq. in.

All these figures for pipe line pressure drop count on practically uniform flow. Experiments show that for rapidly pulsating flow, as for instance air to a rock drill, the pressure drop increases rapidly. At present there is no complete data available to calculate this, because it depends upon many factors, including the rate of pulsation. An air hammer is a much slower operating device than a rock drill; nevertheless, the pressure drop to a hammer would undoubtedly be greater than for steady flow. In such an installation, therefore, it would be advisable, especially with a small pipe line, to place a receiver in the air line just previous to the hammer. This would not be for air storage purposes, but to smooth out the pulsations, thus reducing pipe line friction and maintaining a more constant pressure right at the hammer. In the case in point, the 2-in. air line could connect to a receiver, say 14 in. by 4 ft.

A Single-stage Air Compressor

This brings us to the question of storage capacity obtained by allowing a certain pressure drop in a receiver. These figures are sometimes disappointing, as requiring excessive receiver volume, but to show how it is figured we will give an example. In such a system there would be an appreciable temperature change while filling and emptying the receiver; but as this would seriously complicate the problem, we will omit this effect. In the necessarily approximate nature of such a problem, where so many assumptions of time of operation, pressures in the hammer cylinder, etc., have to be made, this temperature effect may well be neglected.

Receiver Storage Capacity

In the original problem we figured that the two hammers would require a total of 413 cu. ft. of free air per minute. We also calculated that the compressor output was 604 cu. ft. per minute. In most establishments, other machines would be consuming air, so let us assume this other air usage amounts to 300 ft. per minute. The total consumption would then be 713 cu. ft. which exceeds the compressor output by 109 cu. ft. per minute. We will suppose that the maximum hammer consumption lasts for three minutes and that the usage of the rest of the shop remains practically constant at 300 cu. ft. per minute. Also we will assume that both hammers remain idle four minutes after each working period. Also the pressure in the system must not be allowed to drop below 60 lb. What receiver capacity must be provided?

Starting with the pressure at 100 lb., at the end of three minutes the total air consumed by the shop would be

$$3 \times 713 = 2{,}139 \text{ cu. ft.}$$

During that time the compressors will have furnished

$$3 \times 604 = 1{,}812 \text{ cu. ft.}$$

The difference, or 327 cu. ft., must have come from storage.

If we let the receiver capacity be V cu. ft., the amount of *free air* stored in it when the pressure is 100 lb. will be

$$\frac{V \times (100 + 14.7)}{14.7} \text{ cu. ft.}$$

because if all the air inside were expanded to atmosphere (free air), its volume would be increased in the inverse ratio of the absolute pressure change, if we neglect temperature change. This is the equivalent *free* air. Similarly, at the end of the three-minute cycle, when the pressure has dropped to 60 lb., the stored air remaining inside the receiver would be

$$\frac{V \times (60 + 14.7)}{14.7} \text{ cu. ft.}$$

free air; that is, this would be its volume if expanded from 60 lb. to atmosphere, the temperature change being neglected. We then have:

Free air in receiver at beginning of three minutes run.	$V \times \frac{114.7}{14.7}$
Free air in receiver at end of three minutes run.	$V \times \frac{74.7}{14.7}$
Difference equals free air delivered to pipe line by expansion of air in receiver.	$V \times \frac{114.7}{14.7} - V \times \frac{74.7}{14.7}$

This becomes

$$\frac{V \times 114.7 - 74.7}{14.7} = V \times \frac{40}{14.7}$$

or 2.72 V

Our problem states we must furnish a total of 327 cu. ft. of free air by this change of receiver pressure, so we have

$$2.72\,V = 327$$

$$\text{or } V = \frac{327}{2.72} = 120 \text{ cu. ft.}$$

This is the required cubic contents of the receiver. The nearest commercial size would be 48-in. diam. × 12-ft. high, having a volume of 152 cu. ft.

Using this size it would be interesting to figure what the drop in pressure will be after it has furnished 327 cu. ft. of free air by expansion from 100 lb.

As before:

Free air in receiver at start.	$\frac{152 \times 114.7}{14.7} = 1{,}186$ cu. ft.
Deduct 327 ft. delivered to pipe line.	$\frac{327}{}$

Free air left in receiver....859 cu. ft.

To squeeze 859 cu. ft. of free air into a cubic content of 152 ft., the absolute pressure will be increased just as many times as the volume will be decreased (again neglecting temperature change).

Thus the pressure will be

$$\frac{859}{152} \times 14.7 = 83 \text{ lb. absolute.}$$

Deducting 14.7 lb. atmospheric pressure, we obtain

$$83 - 14.7 = 68.3 \text{ lb. gage}$$

as the final pressure in the receiver at the end of the three minute period of maximum demand.

We have stated in the problem that in the next four minutes the hammers are idle; but the rest of the shop will continue to use 300 ft. of free air per minute. The shop consumption during the next four minutes will then be

$$4 \times 300 = 1{,}200 \text{ cu. ft.}$$

and the compressor output will be

$$4 \times 604 = 2{,}416.$$

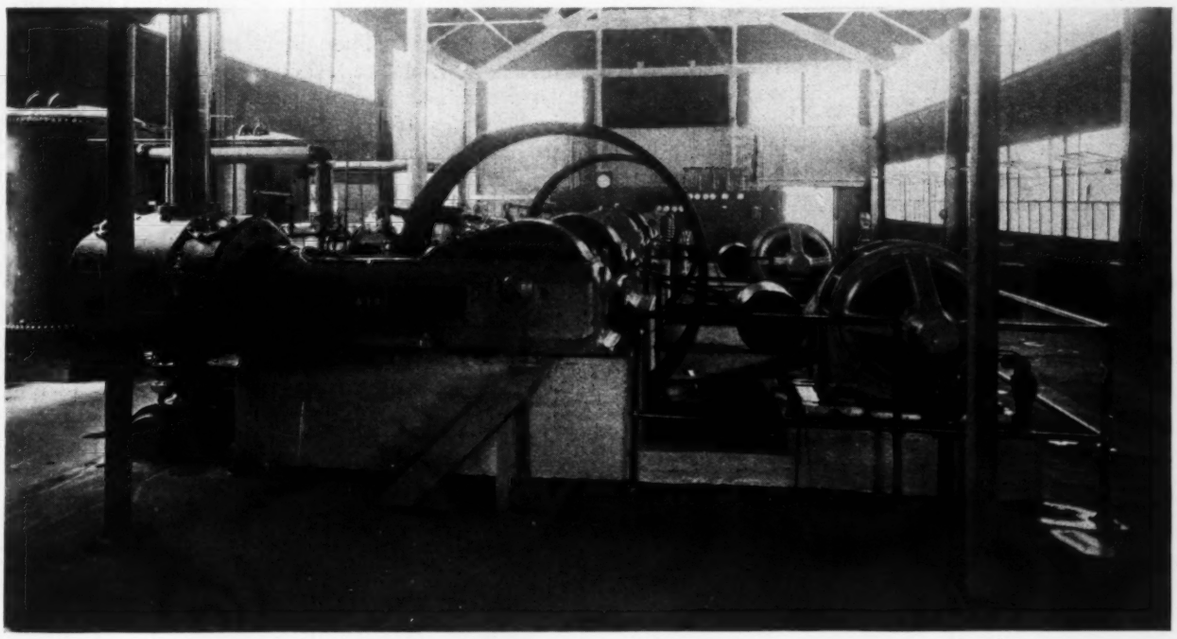
There would be an accumulation of

$$2{,}416 - 1{,}200 = 1{,}216 \text{ cu. ft.}$$

of free air during the four minutes time. It will, therefore, require only

$$\frac{327}{1{,}216} \times 4 = 1.07 \text{ minutes}$$

time to replace the air expanded from the receiver and to bring the pressure back to 100 lb. Therefore, the idle time of the hammers could be reduced almost to one minute and still be able to supply the air. Under the conditions of the problem the unloading mechanism of the compressor would prevent the air pressure from exceeding 100 lbs. and the supply of air would automatically be cut off and the power correspondingly reduced.



A Compound Air Compressor

Pressure Drop—Branched Pipe

This asumption that while the hammers require 413 cu. ft. and the rest of the shop is using 300 cu. ft. per minute, would alter that part of the problem referring to the pressure drop. Doubtless the total quantity of 713 cu. ft. would pass through the 4-in. pipe, while only the 413 cu. ft. would go through the 2-in. pipe. This would also complicate the problem of turning the air in the tee, because only the 413 cu. ft. would make this turn. We then have that 713 cu. ft. are flowing through 65 ft. of 4-in. straight pipe and 413 cu. ft. are flowing through the 30 extra feet of 4-in. pipe which are equivalent to the effect of the reducing tee. We have already figured that 413 cu. ft. of air passing through 65 ft. of 4-in. pipe would have a pressure drop of .0306 lb. per sq. in. Rather than become involved in the squares and 5th powers of the full formula, let us again use proportions. The formula shows that the pressure drop is proportional of the square of the quantity flowing, so we have

$$\frac{.0306 \times 713^2}{413^2} = .0912 \text{ lb. per sq. in.}$$

for 713 cu. ft. through the 65 ft. straight length.

Similarly, by proportion, the drop due to 413 cu. ft. through the 4-in. reducing tee (equivalent to 30 ft. of 4-in. pipe) would be

$$\frac{.0306 \times 30}{65} = .0141 \text{ lb. per sq. in.}$$

The previous calculation for the 413 cu. ft. through the 15 ft. of 2-in. pipe and its globe valve is correct and need not be altered, so adding up all the pressure drops calculated for the various sections of this branching pipe system, we have

	Lb. per sq. in.
713 cu. ft. through 65 ft. 4-in. pipe	= .0912
413 cu. ft. turning the 4-in. reducing tee	= .0141
413 cu. ft. through 15 ft. of straight 2-in. pipe, plus 14½ ft. equivalent to globe valve	= .4660
Total pressure drop	.5713

Procedure in Computing Losses

From the various angles from which we have viewed this question of pressure drop, it should be plain that the procedure is to divide up the pipe system into sections and consider the diameter and length of each section and the quantity flowing in that section.

After obtaining the pressure loss through each section, the sum of all the sections through which the air flows in series is the total drop. If the drop is great enough, the pressure ratio r in the various sections will decrease as the pressure becomes lower due to the drop. However, such calculations can never be accurate, so there is no use introducing too many complications and the suggested methods of using proportionate values from one condition to another will often save a lot of tedious calculations.

Sizing Chimneys for Tall Buildings

By Robert E. Hattis

IT IS very difficult to make an article on chimneys both interesting and enlightening. While they are simple cylinders filled with hot moving gases, it is surprising how complicated a mathematical analysis becomes which covers all governing factors. Yet it is dangerous to take a hit-or-miss chance on sizing a chimney. If a mistake is made, it is one of those things that is just too bad. Hardly anything can be done to make a chimney which is too small work. Adding height sometimes doesn't do any good, because this adds friction as well as draft and the gain is trivial. Making a chimney larger in diameter is practically impossible in most city buildings. So it is evident that proper sizing of chimneys is very important.

The thought occurs to some of us, who want to play it safe and yet not be involved in a complicated mathematical problem, to make the chimney larger than it needs to be. Yet, the writer knows of a case where a greenhouse put in a stack so large that the boiler couldn't heat the chimney and there was no draft at all. Finally, the owner had to put a steel flue inside the larger stack to make his boiler work. Playing it safe didn't do any good there.

A chimney ought to fit its job. If it's too small your plant can't deliver its capacity and you have smoke. If it's too large it may not work at all or may make it very difficult to carry very low loads economically. Furthermore, if the chimney is made too large, both the increased cost of the chimney and the cost of building or space housing it run up and money is wasted. There are cases where an excellent building arrangement can not be used because playing safe with the chimney makes it take too much room to permit it. It is the duty of the engineer planning the mechanical equipment of a building to help make the best arrangements work, if his more intimate knowledge permits condensing the space his equipment requires. However, if the cost of these engineering calculations equals the savings effected, nothing is gained. This article is being written to facilitate quick calculations in the sizing of chimneys for low pressure boiler heating plants in tall buildings in order to save time, space and money.

A chimney ought to fit its job. Proper sizing of chimneys is very important.

In connection with power boilers it is customary to figure a chimney on a basis of a summation of draft loss through the fuel bed, the boiler, the breeching, the bends, the velocity loss and the chimney friction loss and to provide the required height and diameter to overcome these losses and carry the volume of gases produced away.

But with heating boilers it is customary to rely on the stack recommendation of the boiler manufacturer. This recommendation, however, might be for a stack 52 inches in diameter and 100 feet high, whereas the boilers are for a 21-story building requiring a minimum stack height of 250 feet.

How to determine the diameter of the stack of greater height which is equivalent to the shorter and larger diameter stack recommended by the boiler manufacturer?

Mr. Hattis tells how in this article—with charts that make the calculations a matter of seconds.

The Problem—Height and Diameter

Generally the problem is to get enough height for sufficient draft. The tall chimney presents no such problem. With the tall chimney the problem is reversed. How much smaller in diameter can we make the chimney in buildings 15, 20, 30 or more stories high so as not to have too much draft? This is the question to be answered.

In connection with power boilers it is customary to figure a chimney on a basis of a summation of draft loss through the fuel bed, the boiler, the breeching, the bends, the velocity loss and the chimney friction loss, providing the required height and diameter to overcome those losses and to carry the volume of gases produced away. If we were to total the draft losses from the ash pit to the point of entrance of the breeching into the stack, we have what is termed the "required draft." To provide this required draft, we have a chimney of a certain height and diameter to make available at the breeching connection an "available draft" equal to this required draft. This available draft is equal to the difference between the "theoretical draft," which is based on the height, and the sum of the "chimney friction" and "velocity head" losses which are based on diameter and height.

However, with heating boilers it is customary to rely on the stack recommendation of the boiler manufacturer. He tells us that a chimney for one or two boilers should be a certain minimum height and a certain diameter to correspond with that height. This is in effect a statement signifying that with the draft "available" from the chimney recommended, the draft "required" by the boilers under the usual operating conditions and with a short straight breeching will be equal, and the boilers will deliver their rated loads. In fact, most boiler manufacturers' chimney recommendations will take care of from 150 per cent to 200 per cent of boiler rating. The writer is in favor of using not less than the chimney

sizes recommended by reliable boiler manufacturers for hand fired coal burning boilers.

Making Tall Stack Equivalent to Short

As we are limiting our discussion to chimneys for low pressure boiler heating plants, in tall buildings, we fall in line with the usual custom and look for our stack in the boiler manufacturer's catalog. Our building is, let us say, twenty-one stories and basement high, with elevator penthouses making our minimum stack height 250 feet, and yet the boiler man says his stack shall be 52-in. in diameter and 100 ft. high. As it is customary in these tall buildings, which usually are apartment, loft or office buildings, to build these stacks of steel and as it is necessary to line the stacks part way up, at least, with 2 in. of lining, the inside diameter of the steel stack itself is 56-in. If this lining has a finish coat of ½ in. of cement, the stack is 57 in. inside of steel. The stack has at least ⅜-in. thickness at the bottom which makes the outside diameter 57¾ in. Add to this 5 in. for a 2½-in. bolting angle at the joints and we have 62¾ in.

If this stack goes inside the building, provide 1 in. for clearance and you have a square space 63¾ in. or 64 in. This must be housed in not less than 3-in. tile wall with ¾-in. of plaster which means 71½ in. or roughly 6 feet square of space through the entire building from basement through the roof required by the stack. The volume of the space taken by the stack is, allowing for ten feet projection about the roof, 8,640 cu. ft. This at 60 cts. per cu. ft. equals $5,184 plus the cost of the stack.

If you put the stack out-of-doors you have a perpetual painting, obstructions to natural light in adjacent windows, obstructions to passage-ways below. Certainly the smaller this obstruction the better; the less space occupied the less costly the space used by stack; the smaller the stack diameter the less the stack cost. Isn't it worth while to make the stack as small in diameter as the increased height will enable you to and yet have a stack equivalent to that recommended by the boiler manufacturer?

Charts Work It Out

The following set of curves enables you to obtain the diameter of the stack of greater height which is equiva-

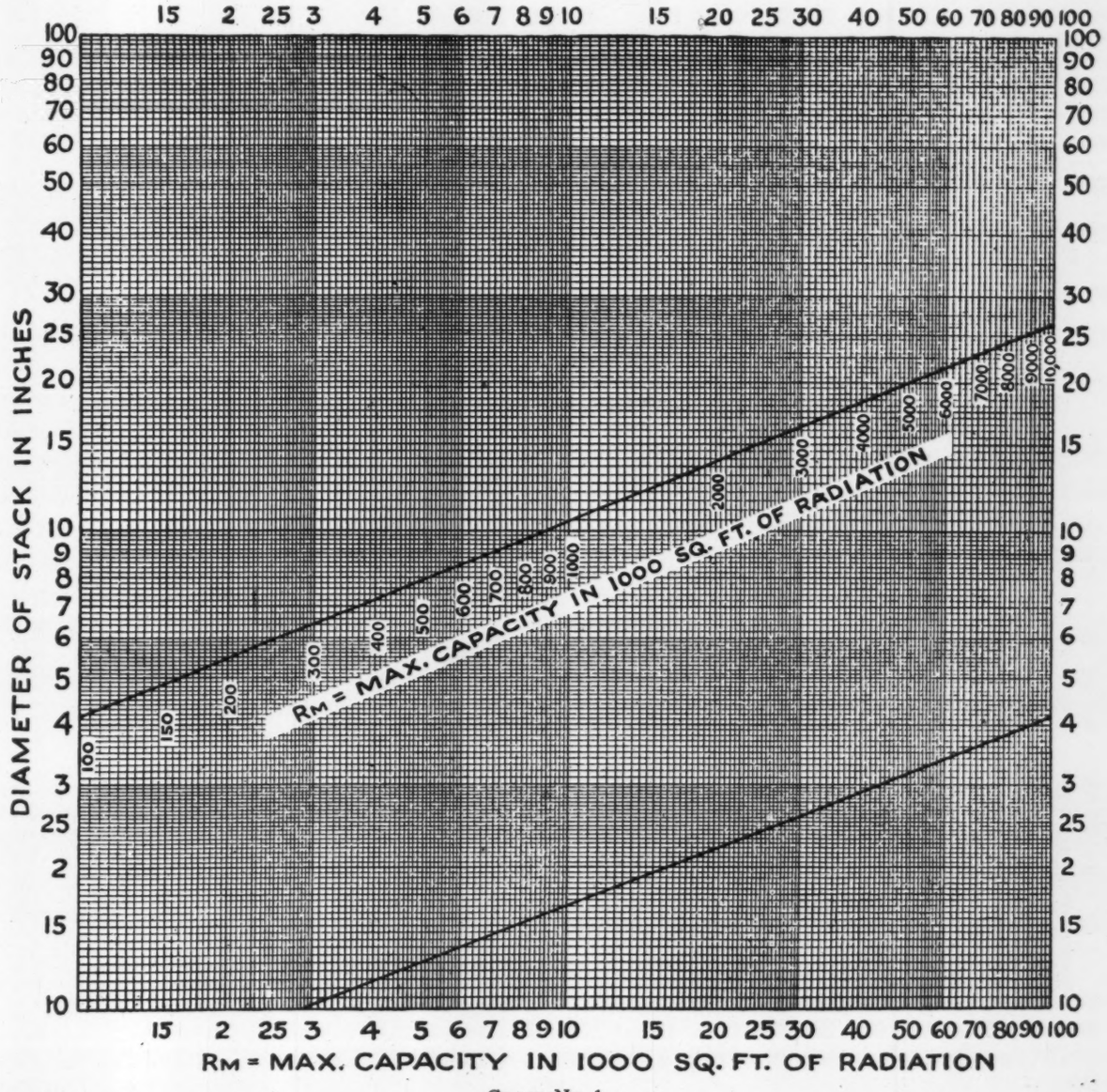


CHART No. 1

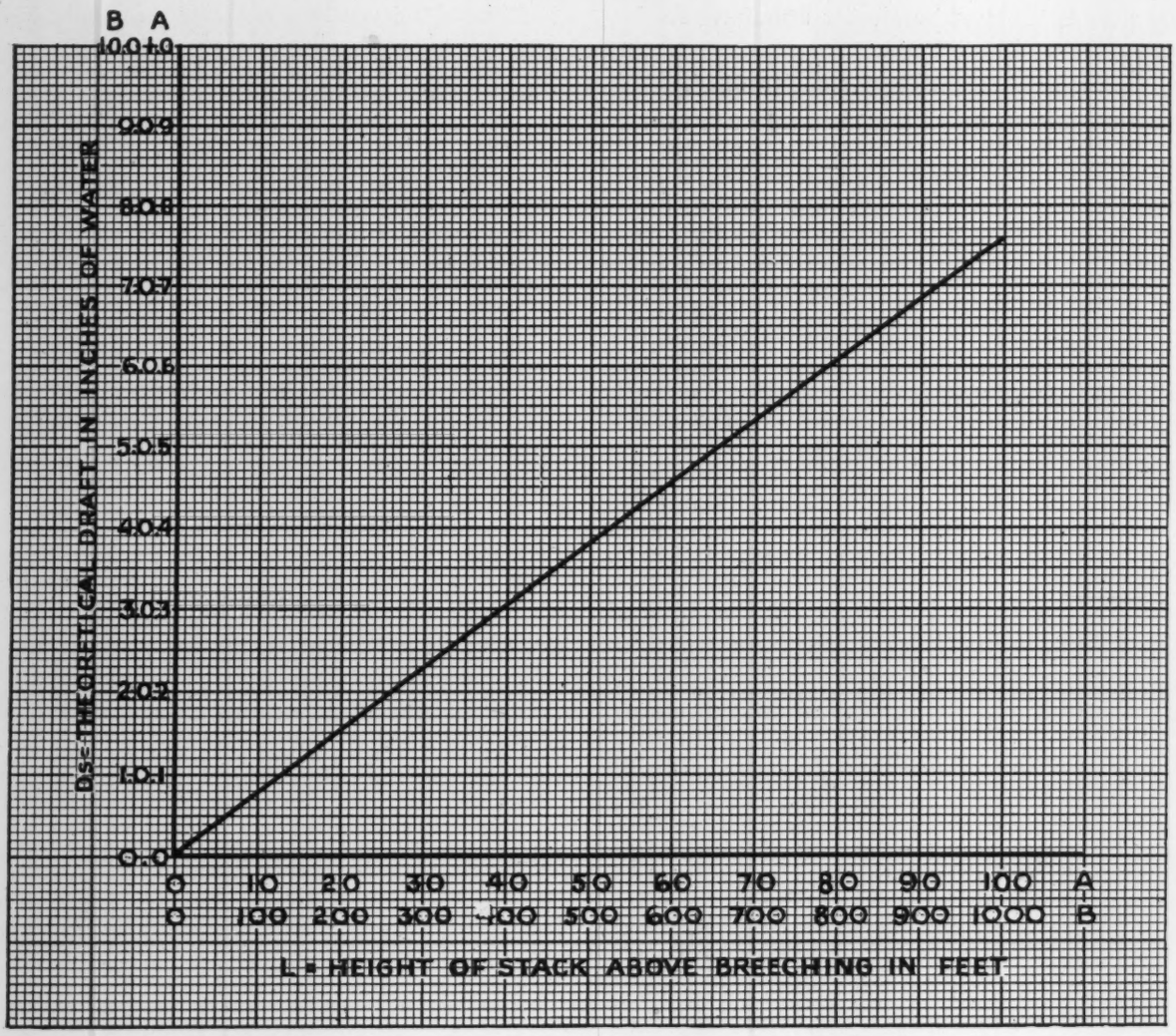


CHART No. 2

lent to the shorter and larger diameter stack given in the boiler manufacturer's catalog. These are based on average heating boiler operating characteristics which are taken as follows:

Average stack temperature.....500 deg. fahr.
Outside air temperature..... 32 deg. fahr.
Boiler efficiency.....70 per cent
Combustion gases per pound of coal..17.25 lbs.
Overload capacity provided for.....50 per cent
Altitude.....Sea Level

Chart No. 1 shows the relation between *maximum stack capacity* in thousand square feet of radiation and *diameter* of stack in inches. Allowance is made for 50 per cent additional capacity above that shown. The formula for the curve drawn is $R_m = 9.17\ D^{5/2}$.

Chart No. 2 shows the relation between *theoretical draft* in inches of water to stack *height* in feet. The formula for the curve drawn is $D_s = .0076\ L$. "L" is the stack height above the breeching. D_s is theoretical static draft.

Chart No. 3 shows the relation between the available draft fraction and the fraction of maximum capacity. The writer is indebted to an American Society of Mechanical Engineers Bulletin entitled "Chimney Sizes," written by Alfred Cotton in May, 1923. The formula for this curve is $\frac{D_a}{D_s} = 1 - \left(\frac{R}{R_m}\right)^2$. D_a is available draft at base of stack. R = sum of boiler ratings in square feet. R_m = maximum capacity of stack in square feet of radiation.

The following instructions show how to use the charts:

Low Stack

Given "L" in ft., "D" in inches and "R" in sq. ft. of Radiation. Find "D_a".

Step 1. With D on Chart 1, get R_m.

Step 2. With R and R_m calculate $\frac{R}{R_m}$ on slide rule.

Step 3. With $\frac{R}{R_m}$ on Chart 3, get $\frac{D_a}{D_s}$.

Step 4. With L on Chart 2, get D_s.

Step 5. With $\frac{D_a}{D_s}$ and D_s, get D_a, which is equal to $\frac{D_a}{D_s} \times D_s$.

High Stack

Given L in feet and R in sq. ft. of radiation.

Since same boilers and breeching are used, "D_a" for low stack equals "D_a" for high stack. Then we have L, R and D_a.

Step 1. With L on Chart 2, get D_s.

Step 2. With D_a and D_s, calculate $\frac{D_a}{D_s}$ on slide rule.

Step 3. With $\frac{D_a}{D_s}$ on Chart 3, get $\frac{R}{R_m}$.

Step 4. With R and $\frac{R}{R_m}$ get R_m, which is equal to R divided by $\frac{R}{R_m}$.

Step 5. With R_m on Chart 1, get D.

The following tabulation is recommended for facilitating the use of the charts:

Stack	R	L	D	R_m	R/R_m	D_a/D_s	D_s	D_a
Low	44,000	100	52″	178,000	.247	.936	.76	.711
High	44,000	250	33″	55,600	.792	.374	1.9	.711

Example

Take the example referred to earlier in the article. For two 22,000 sq. ft. boilers, boiler catalog requires 52-in. stack 100-ft. high. What is the diameter of a 250-ft. stack which will give the same available draft as that of the stack required by the boiler catalog?

Write in the known quantities for the low stack, which are L, D and R. With Step 1 get R_m for 52-in. stack, which is equal to 178,000. With Step 2 and slide rule, $\frac{R}{R_m}$ equals .247.

With Step 3 and R/R_m, $\frac{D_a}{D_s}$ equals .936. With Step 4, and L equal 100 ft., find D_s equal to .76. With Step 5 $\frac{D_a}{D_s} \times D_s$ equals $.936 \times .76 = .711$.

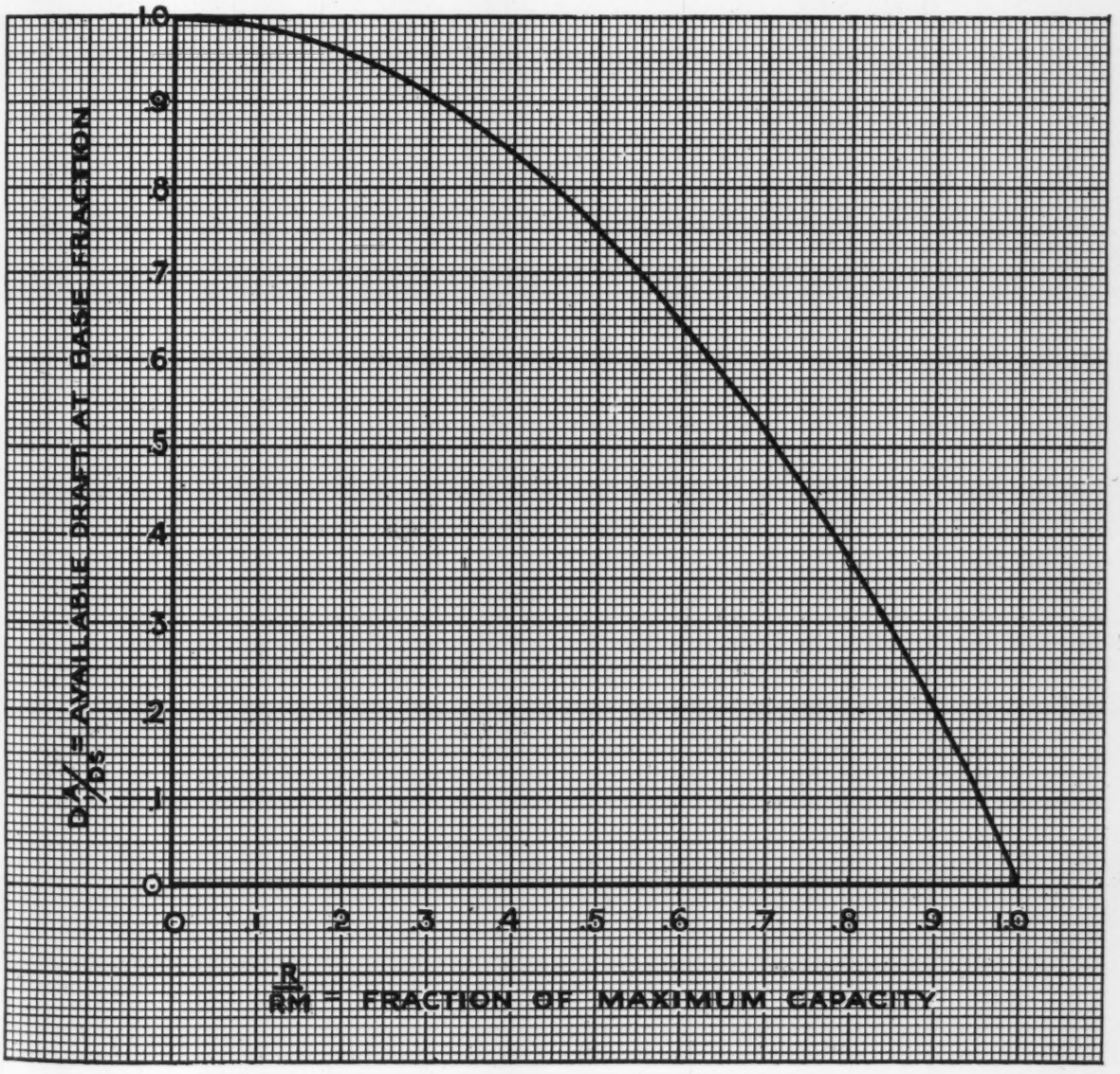


CHART NO. 3

For the high stack, write in L and R which are known, and also D_a which has just been calculated for the low stack. With Step 1 and L = 250 ft., get D_s which is equal to 1.9. With Step 2 $\frac{D_a}{D_s} = \frac{.711}{1.9} = .374$. With Step 3, with $\frac{D_a}{D_s} = .374$, get $\frac{R}{R_m} = .792$.

With step 4 having R and R/R_m, $\frac{R}{\left(\frac{R}{R_m}\right)} = \frac{44{,}000}{.792} =$ 55,600. With step 5 and R_m, get "D" = 33 in., which is what we are striving to obtain.

Savings Effected

Let us see what, in addition to other advantages, we gain in dollars and cents by using the 33 in. stack instead of the 52 in. we would have used without calculation. We learned previously that a 52 in. stack takes up a space 72 in. square or a space 20 in. greater than the stack diameter. With a 33 in. stack, the space required is 33 in. + 20 in. = 53 in. square. The cost of the space required by the 33 in. stack is $\left(\frac{53}{72}\right)^2 \times \$5184 =$ $2810. This is a saving of $2374 in cost of space plus the saving in cost of a smaller stack.

Isn't it worthwhile capitalizing on the height of the stack to cut down on its diameter?

Low Humidity Drying

By Malcolm C. W. Tomlinson

ONCE the decision is made to investigate the costs and, therefore, the advisability of an investment in low humidity air conditioning equipment the first move should be the determination of the heat balance for the room, or rooms, which it is proposed to air condition. A survey showing the various heat losses and heat gains for the extremes and the mean weather conditions will likely show weaknesses in construction details not heretofore suspected and will be of immense help to the refrigeration and air conditioning firms who may be called upon to figure the proposition. Very valuable data on heat transmission, infiltration, insulation, etc., can be secured from the American Society of Refrigerating Engineers and also from the American Society of Heating & Ventilating Engineers.

The Cooling Equipment

Probably the next step will be to determine the character or type of cooling equipment that will be used. The most simple proposition, but not always the wisest plan, is the use of water from deep wells or nearby rivers. The difficulties come in the variation of the temperature and also in the amount of water which must be handled to obtain a comparatively small degree of cooling. Small installations may be handled by chemical means such as silica gel. The latter is a crystalline structure which has a remarkable affinity for water. It is usually provided in at least two containers, or tanks, provided with interchangeable valve control. This makes it possible to activate one tank while the other tank is in service. Activation is the process of driving off the moisture by means of high temperatures. Silica gel loses its efficiency in the presence of carbon compounds and, for this reason, should not be activated by gas oil unless special protection is afforded against the products of combustion. It is most efficient in moisture removal when the air-water vapor mixtures are cool. It is not, therefore, a refrigerating or cooling medium but, as an auxiliary to refrigerating equipment where large quantities of moisture must be handled, it has considerable usefulness. Two outstanding types of refrigerating equipment on the market today are the carbon dioxide and the diolene. The first is reciprocating and operates at high pressures while the second is of the centrifugal type and operates under a vacuum.

Cooling mediums available include sodium chloride, calcium chloride and ethylene glycol. The first is quite corrosive, the second entrains heavily in the air when used with air washers while the third is expensive and must be provided with an evaporator for the removal of moisture, or excess water, in order to keep it at the required strength.

Duct Work and Washers

Duct work for air movement should provide a return system where a balanced condition is desired. A tendency in duct design today is towards high velocities in order that the air ducts will not occupy too much space.

Air washers and filters are also necessary adjunct equipment for air conditioning. The washers are especially valuable in removal of dust and dirt while various types of filters have been devised, in recent years, which also are highly efficient in this respect. The washers provide an additional feature in that they are also used for cooling purposes. The removal of fumes is not possible with these forms of equipment.

Infiltration

For low humidity work it is essential that consideration be given to infiltration of air and moisture. Doors leading to rooms with higher humidities should be provided with corridors and air locked so that only one door at a time can be open in each corridor. Walls, floors and ceilings must be waterproofed exceedingly well. Materials used for construction work can be selected to aid the waterproofing process and brick or tile walls can be protected to a considerable degree by hard wall plaster.

Determining the Capacity of a Humidifying Plant

By A. W. Thompson

THE determination of proper capacity for humidifying or air conditioning equipment is a rather nice problem in thermal engineering. Correct solution requires the collection of accurate data in regard to all local conditions which will affect capacity requirements. Upon such fundamental data, capacity may be calculated with a degree of accuracy which will insure satisfactory performance when equipment with corresponding adequacy has been installed.

The method considered as a whole is a practical application of the laws of physics and thermodynamics which have been taught in courses on heating and ventilation at the engineering schools for a generation past. Such methods have always been used in the design of successful central station humidifying systems because of the importance of having the ventilating fans and ducts and the humidifying and heating accessories of adequate capacity to meet the required conditions.

The self same methods should always be applied in the design of a system of distributed humidifiers, such as the atomizer or high duty types. Systems of distributed humidifiers can be adapted to almost any building construction and are widely used in industries where the practice of humidification is common, and particularly in the textile industry. It is an unfortunate fact, however, that the importance of capacity and correct methods for determining it have received scant attention in the case of distributed humidifier systems, and it would seem as if the laws which should govern their use are honored much more frequently in their breach than in their observance.

It is, therefore, my purpose to describe correct methods, to explain their vital importance, and to illustrate their proper application in some simple typical instances.

Humidifying a Typical Textile Factory

Let us first consider the case of a typical textile factory where artificial humidity is required throughout all parts of the year because the atmospheric moisture provided by nature is insufficient for the best processing of textiles.

Humidifiers will be in use the year round, but the extent to which they are used from day to day and hour to hour will fluctuate and vary widely because fluctuating weather conditions cause corresponding fluctuations in the "demand for moisture." How then shall we determine those conditions which will develop our peak load and how determine its magnitude?

Air having a variable initial moisture content, depending upon the prevailing weather, will pass through every department in the factory as the result of either natural or artificial ventilation. We must charge this air with water vapor in sufficient quantity to produce that level of relative humidity which we desire, thus making up the "moisture deficit" represented by the difference between the absolute humidity of the outside air and the absolute humidity of the air inside at its ultimate condition of temperature and humidity.

Temperature an Important Factor

Temperature is an important factor, and in hot summer weather we shall have to deal with liberal rates of ventilation. Evaporation must be sufficient to maintain the humidity and also effect a sufficient cooling of the air to maintain a favorable "effective temperature"—that is to say, a relation between humidity and dry bulb temperature which will insure reasonable comfort for operatives so that their industrial efficiency shall not be impaired by an oppressive atmosphere.

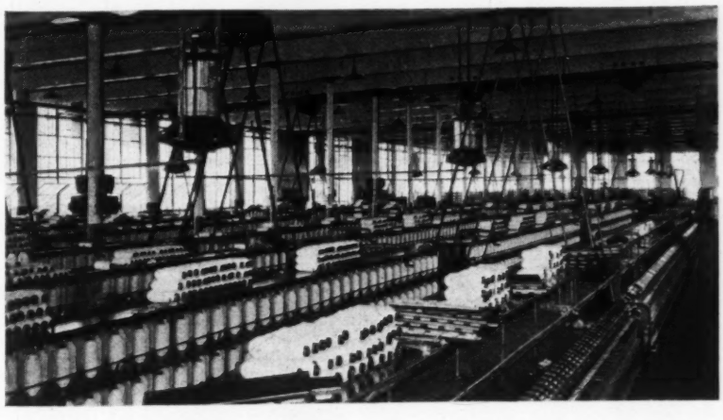


FIG 1.—A COTTON SPINNING ROOM. DEPARTMENTS WHERE A LARGE AMOUNT OF HEAT IS LIBERATED REQUIRE HIGH HUMIDIFYING CAPACITY

Determine Rate of Heat Liberation

Assurance of the humidity desired coupled with tolerable temperatures will depend chiefly upon our success in proportioning the amount of water evaporated to the rate of heat liberation with which we have to contend.

We must, therefore, determine with especial care the rate at which heat will be dissipated by the consumption of power; by infiltration through roof, side walls and windows; and by the bodily heat emanation of operatives.

Power consumption and heat infiltration are especially important factors in textile mills, but the effect of bodily emanation is usually so small as to be negligible except in industries where departments are relatively crowded with operatives.

In determining the rate of heat dissipation by power consumption, we should deal separately with each important group of productive machinery. In a large spinning room it is not uncommon to find warp spinning, filling spinning, and spooling and warping processes conducted on the same floor. In such cases the power consumption should be separately determined for each of the various areas over which the different processes are conducted.

Total electric input to each important area should be carefully estimated. Conductor and motor losses, as well as actual power input to productive machinery, are represented by equivalent liberation of heat.

In estimating heat infiltration, it should be noted that roof surfaces exposed to the direct rays of the sun will take on temperatures which are much higher than the temperature of the outside air. The difference between outside and inside roof temperatures represents the "thermal head" which will drive heat inward at a rate depending upon the thickness and the properties of the particular material of which the roof is constructed. Infiltration through side walls will depend upon similar considerations. Sunlight entering through windows unprotected by curtains or awnings is converted into a considerable amount of heat and should be represented by a proper allowance.

Normal Weather Extreme

In winter the demand for moisture is relatively small because of the then prevailing low rates of ventilation. The peak load will occur in hot, dry summer weather when rates of ventilation are certain to be relatively high.

It becomes important, then, to determine those extremes of temperature and humidity which will occur with reasonable frequency during summer in the locality under consideration. We must fix upon that combination of high temperature, coupled with low relative humidity which corresponds to the "normal weather extreme," for it is this condition which will develop the maximum demand for moisture, or peak load.

The United States weather bureau maintains numerous observation stations where temperature and relative humidity are observed at morning, noon and night. The records of neighboring stations can always be procured and should be used to establish the local weather expectancy. Noon observations are most important but it should be remembered that the maximum temperature for the day is usually reached in mid-afternoon and commonly exceeds the noon temperature by some 3 or 4 degrees.

Analysis of weather bureau reports for several consecutive years justifies the conclusion that the "normal extreme" for New England locations corresponds very closely to a dry bulb temperature of 92 degrees with a wet bulb depression of 20 degrees. In some parts of the South, higher temperatures and greater depressions prevail. The district including most of Georgia, Alabama and eastern Mississisippi in particular, attains a normal extreme approaching more closely to 100 deg. dry bulb temperature at a wet bulb depression of 28 deg.

"Normal extreme," as the term is here used, denotes a condition likely to occur with reasonable frequency for several hours at a time during the usual hot days of any normal summer. The normal extreme is, therefore, in no sense an absolute extreme. Not only is it certain to be exceeded in record-breaking hot weather, but it will be often reached and not infrequently exceeded in the normal hot weather of nearly every year.

It is nevertheless the best commercial basis for the determination of capacity. If we base capacity upon those higher extremes which are attained rather rarely and for comparatively brief intervals, the additional investment in equipment required for capacity corresponding would be unwarranted for its use would be so infrequent that it could earn no adequate return.

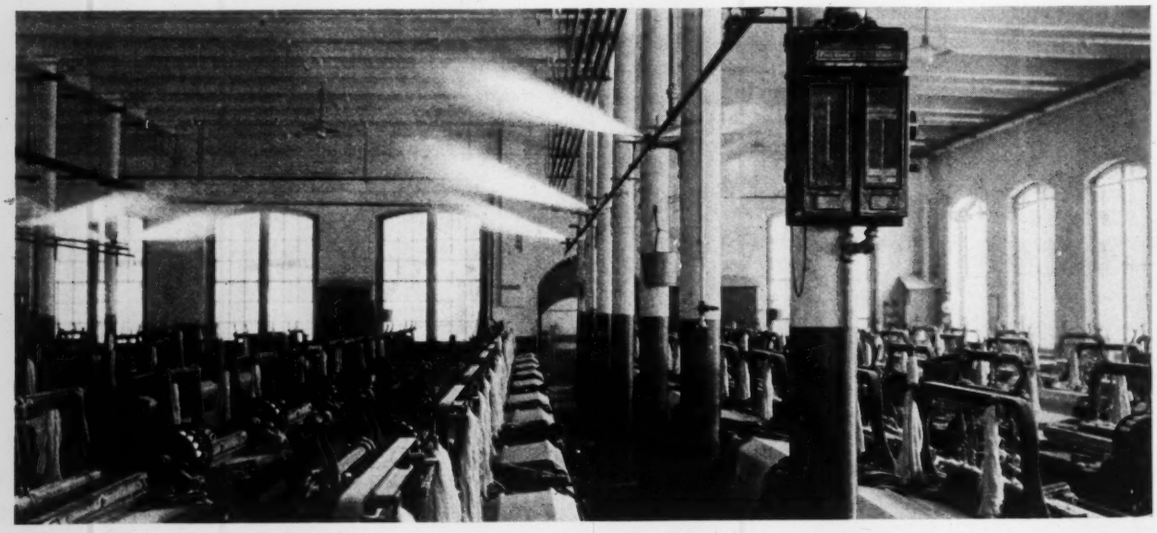


FIG. 2.—HUMIDIFIERS PRODUCING HIGH RELATIVE HUMIDITY IN MODERN COTTON WEAVE ROOM

The Cooling Margin

The next consideration is "cooling effect." How far shall we carry our attempt to keep the temperature down to reasonable levels? Given a stated normal extreme of outside temperature and humidity, what should our room temperature be when this particular condition prevails? The difference between the dry bulb temperature inside the factory and the outside wet bulb temperature is a factor known to humidifying engineers as the "cooling margin." The more we reduce the cooling margin, the more rapid must be the ventilation and the higher the rate at which water must be evaporated.

In determining a proper value for the cooling margin, we should aim to limit the effective temperature to about 82 deg. fahr., although this value is considerably above the so-called "comfort zone." Absolute comfort is an ideal condition and industrial operatives no more expect to enjoy it than do those of us who cheerfully pursue our daily tasks and duties during hot summer weather in spite of some inevitable temporary discomfort.

Operative Efficiency

Research and experience have shown that industrial operatives will perform their tasks efficiently and cheerfully until the effective temperature exceeds 82 deg. Beyond this point discomfort and languor will reduce operative efficiency very appreciably. Humidifying capacity which is sufficient to keep effective temperature below this limit will earn good returns on the investment by insuring continuous operation at good production throughout all ordinary periods of hot weather. If capacity is so inadequate that effective temperature exceeds this limit in ordinary warm weather, operatives will lose in productive efficiency and will insist upon opening transoms so freely for relief that the required humidity will be lost. Bad processing will result during such periods, thus materially reducing production by mechanical as well as human losses in efficiency.

A Typical Problem

With these considerations in mind, let us take up a typical problem. We require the evaporative capacity for a weaving department whose total cubical content is 100,000 cubic feet, within which there is a total heat liberation estimated at 225,000 B.t.u. per hr.

It is required to maintain 85 per cent relative humidity against an outside normal extreme of 92 deg. dry bulb temperature, the wet bulb temperature being 72 deg., corresponding to a wet bulb depression of 20 deg. Referring to proper tables or charts for the relation between effective temperature, dry bulb temperature and humidity,* we shall find that at a relative humidity of 85 per cent we may limit the effective temperature to 82 deg. by limiting the dry bulb temperature to 84 deg.

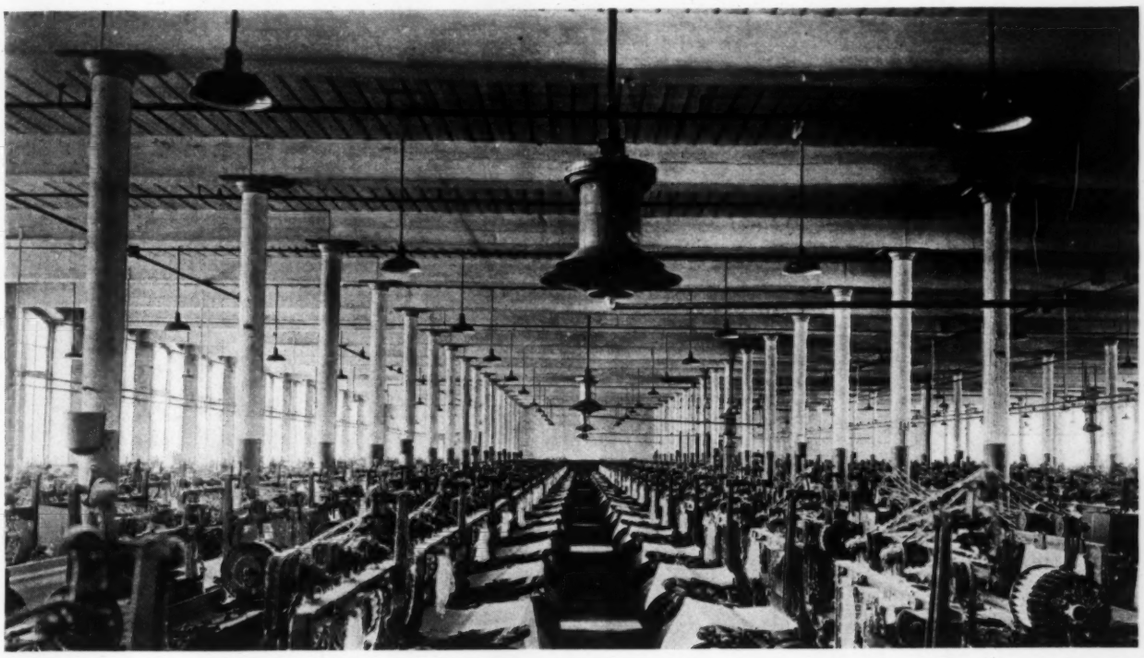


FIG. 3.—ANOTHER COTTON WEAVE ROOM SHOWING HOW HUMIDIFIERS ARE APPLIED

The outside wet bulb temperature being 72 deg., our inside temperature of 84 deg. as thus determined demands a cooling margin of 12 deg.

All of our required data are now at hand, including the predetermined atmospheric and thermal conditions and the final atmospheric conditions prescribed for performance, and we have determined from these the changes which we are required to effect in temperature and moisture content of the air.

Recapitulation

	Outside	Inside	Change in Temperature
Dry bulb temperature, degrees	92	84	—8
Wet bulb temperature, degrees	72	80½	
Relative humidity, per cent	37	85	
Corresponding absolute humidity	5.9 gr.	10.9 gr.	

Moisture deficit = 5.0 grains per cubic foot.
Total additive heat 225,000 B.t.u. per hour.

There now remains the thermal determination of the amount of heat interchanged by virtue of the modification in the air temperature and the vaporization of the

* See Guide of the American Society of Heating and Ventilating Engineers. Also, "Fundamental Laws of Humidity with Charts and Tables"—Bulletin No. 329 published by Parks-Cramer Co.

water required to overcome the moisture deficit; the calculation of the rate of air change which will effect balance between liberation and absorption of heat; and the rate of total evaporation corresponding. By this quite usual method of determination by "Heat Balance" it can be shown that we require in this particular case a total hourly evaporation of 265 pounds of water at a rate of ventilation corresponding to four air changes per hour.

That is to say, in the presence of the prescribed rate of

FIG. 4.—APPLICATION OF AIR CONDITIONING TO A TOBACCO FACTORY

heat liberation, the evaporation of this amount of water into the corresponding volume of air will effect exactly that change in the moisture content and temperature of the air by which it is desired that the final condition should differ from the prescribed initial condition—a matter which is subject to ready proof by computation.

The preceding problem is fairly representative of the requirements of a weaving department in a factory of modern construction located in New England. If the same factory were located in central Alabama, with an outside temperature of 100 deg. fahr. at a wet bulb temperature of 72 deg., we should have a considerably greater heat infiltration and the total heat liberation might easily reach 255,000 B.t.u. per hr. With the same inside atmospheric conditions required, calculation will indicate

FIG. 5. — HUMIDIFICATION APPLIED TO SHOE MULLING

an evaporation of 358 pounds per hour—an increase of 35 per cent above the first instance—at an air change of 4.4 per hour.

As a further illustration of wide variation in capacity requirements, let us compare two spinning rooms of identical atmospheric requirements—say a relative humidity of 65 per cent at 88 deg. fahr. with a cooling margin of 16 deg.

Spinning room *A*, located in New England for the production of fine yarns, has 125 h.p. concentrated in each 100,000 cubic feet, and a total heat liberation of 360,000 B.t.u. per hr., with outside conditions at 92 deg. dry bulb and 20 deg. depression. Calculation indicates an evaporation of 396 pounds of water per hour at 8.6 air changes.

Spinning room *B*, making hard-twisted warp yarns and located in Alabama, consumes 180 h.p. in each 100,000 cubic feet and has a total heat liberation of 512,000 B.t.u., with outside conditions of 100 deg. dry bulb temperature at 28 deg. depression. Calculation indicates 723 pounds of water at 11.9 air changes—a capacity requirement exceeding that of spinning room *A* by no less than 83 per cent.

Not only are just such widely varying requirements as the above met in actual practice, but it is a fact that capacity, as determined by the methods illustrated here to comply with such requirements, has proved to be ample, but no more than adequate—that is, the supply has proved to be well adapted to the demand and the capacity is found to be "commercially good." It is true that in the final instance of spinning room *B*, the requirements as here described are severe but they are not by any means uncommon. In such a case it is absolutely necessary to provide corresponding capacity for it will be impossible otherwise to maintain both humidity and reasonable temperature during hot weather so as to insure good processing and productive efficiency. Only the most efficient types of apparatus can be used successfully to deliver the capacity which is essential for satisfactory results in such cases.

A Paper-Printing Factory

Let us now consider the requirements of a paper-printing factory where power consumption is not so high as to affect temperature very materially. Natural conditions in the summer months are generally satisfactory as to

temperature and humidity, for in this case good processing requires only about 55 per cent relative humidity to prevent the troublesome effect of static electricity. Such a factory may operate from late spring to early fall with little or no need for humidifier operation, but in cold weather, when ventilation is restricted and artificial heating is necessary, humidifiers will be more or less constantly in operation.

We need here only provide sufficient evaporation to

compensate for the deficit in moisture which develops when we admit cold outside air at winter temperatures and low absolute humidity and raise it to from 70 deg. to 75 deg., adding enough moisture to fix the humidity at 55 per cent. Ventilation is not likely to exceed a rate of, say, two air renewals per hour.

Capacity calculation in such a case is obviously simple, and this illustration is given merely to emphasize the requirements of a case which locates the maximum demand for moisture under winter conditions in contra-distinction to the summer requirements of a textile factory.

FIG. 6.—CONTROLLED AIR CONDITIONING IN A PRINTING PLANT.

Summary

In scrutinizing the merits of this method it will at once appear that humidity and temperature, being prerequisites, are arbitrarily determinable, while weather conditions may be easily determined by investigation. Heat liberation, while not a determinable fact, is capable of estimation with reasonably close accuracy. Rates of air change are subject to absolute control by forced ventilation, and in the case of natural ventilation can always be limited to rates which are not excessive. The variability of air change when ventilation is natural in no way reflects unfavorably upon the value of this method of determining capacity.

It is appropriate at this point to reiterate that it is of the utmost importance to determine the heat liberation, the prevailing normal weather extreme and cooling margin with the utmost care. If these factors are not adequately appraised and capacity provided to correspond, the equipment will fail to meet humidity requirements whenever the demand for moisture approaches its peak.

It is true that humidity can not be maintained against excessive ventilation even when adequate evaporative capacity is present, but the fault in that case is due not to lack of equipment but to improper use. This particular difficulty may always be remedied by using ordinary care and common sense to prevent the excessive opening of window transoms.

On the other hand, restricted ventilation can not possibly remedy the bad results which inevitably follow from shortage of capacity. An attempt to restrict ventilation during hot weather in the effort to maintain proper humidity with inadequate capacity will result in a rise of temperature to such intolerably high levels that increased ventilation at the expense of humidity is the only recourse, and processing and production will have to suffer.

Peak Load Important

Guess-work and rule-of-thumb methods are entirely out of place in determining proper capacity for humidifying equipment of any type whatever. Uniform conditions can only be secured when sufficient surplus capacity is available for the control of humidity by automatic regulation. Humidifying equipment which can not carry its ordinary peak load successfully will reflect little credit upon its designer and will be as unsatisfactory to its owner as an overloaded steam engine.

The real value of humidifying equipment depends upon its possessing such adequate evaporative capacity that it will perform satisfactorily under its peak load. It, therefore, behooves us to use the utmost care in determining the peak load, and then to make certain that the equipment to be actually provided can not fail to deliver the corresponding evaporation.

Goodenough, International Authority on Thermodynamics, Dies

Prof. George A. Goodenough, engineer of international repute and professor of themodynamics at the University of Illinois, died at his home in Urbana, September 29.

In the death of Goodenough, the engineering profession has lost an outstanding leader. He was prominent also in educational and athletic circles, serving as chairman of the western conference faculty committee on athletics. He had been a member of the University faculty since 1899, and was considered one of the most successful teachers of thermodynamics.

Professor Goodenough's engineering achievements were numerous. His steam tables are considered authoritative throughout the entire world and his work is considered perhaps the most reliable yet done in investigating the properties of steam. An interesting story is told of the manner in which Professor Goodenough read the publisher's proof when his steam tables were being prepared for presentation to the engineering profession. According to his associates, the original manuscript was destroyed, and the thousands of values were re-computed in order to be sure that no errors had crept in during the preparation of the manuscript or the type setting.

"Principles of Thermodynamics" and the chapter on that subject in the handbook of the A. S. C. E. were also Professor Goodenough's work. He was co-author of two text-books in calculus recognized as standard in many colleges and universities. For many years he had served as consulting engineer on steam turbines for a large manufacturing company in the United States. A widow and one son survives him.

Economical Pipe Combinations in Heating Systems

By W. F. Schaphorst

THE selection of pipe sizes has always been more or less of a conundrum, even to trained engineers, because so many factors are involved. However, we find that by attacking the problem "backwards" as outlined below, very satisfactory results are usually obtained. We begin at the farthest outlet and work back toward the source of steam supply. There is nothing strictly new about this method, perhaps, but we believe we have simplified the operations considerably.

Figs. 1 & 2 will be found convenient for quickly determining the standard pipe size leading to a definite radiating area. These curves are based on a loss in pressure of one pound when supplying steam to a radiating surface operating under one lb. gage pressure, the temperature of the room being 70 deg. fahr.

Where the pressure in the radiators is less than or greater than one lb. the following table of factors (Table 1) must be applied in conjunction with Table 2.

Table 1

Pressure, lb.	0.0	1	2	3	4	5	6	8	10
Factor	1.025	1.00	.985	.965	.935	.928	.915	.885	.860

Table 2

Press. loss, lb	.....	0.25	0.50	1.00	1.50	2.00	3.00	4.00	5.00
Factor	.....	0.50	0.707	1.00	1.22	1.41	1.73	2.00	2.23

The application of these tables is very simple and will be explained in examples that follow.

How to Use the Charts

Four general rules, or steps, may be laid down, which, if followed, will almost invariably lead to the correct answer in the determination of pipe sizes by the use of these charts and tables. The rules are:

(1) Enter the right-hand scale of Figs. 1 and 2 with the square feet of radiation to be supplied.

(2) Proceed horizontally to the vertical line running down from the length of the main.

(3) From this intersection proceed downward to the first curve. The size of pipe required is given at the lower end of the curve.

(4) Where either the pressure supplied the radiators or the pressure loss in the main is more or less than 1 lb., enter the right-hand scale with the sq. ft. of radiation + (the factor given in Table 1, under the pressure supplied × the factor given in Table 2, under the pressure loss), and proceed as in the above paragraphs 2 and 3.

An Example

To make the above perfectly clear we will take the following example:

Length of main, 250 ft.
Radiating surface, 9,500 sq. ft.
Steam pressure supplied to radiators, 2 lb.
Pressure loss, 0.5 lb.

9,500 + (.985 × .707) = 13,644. Enter chart, Fig. 2, at 13,644 sq. ft. and find intersection with 250 ft. at left. The first curve below shows that a 6 in. pipe is required.

The above method will take care of the majority of cases that come before the average heating and piping engineer. We believe it to be simpler than methods in general use, and being unusually accurate, we feel that it will prove of considerable value to all who choose to use it.

Further, this method is applicable to the more complicated problems as well as the simpler ones such as have just been cited. We will apply it to a

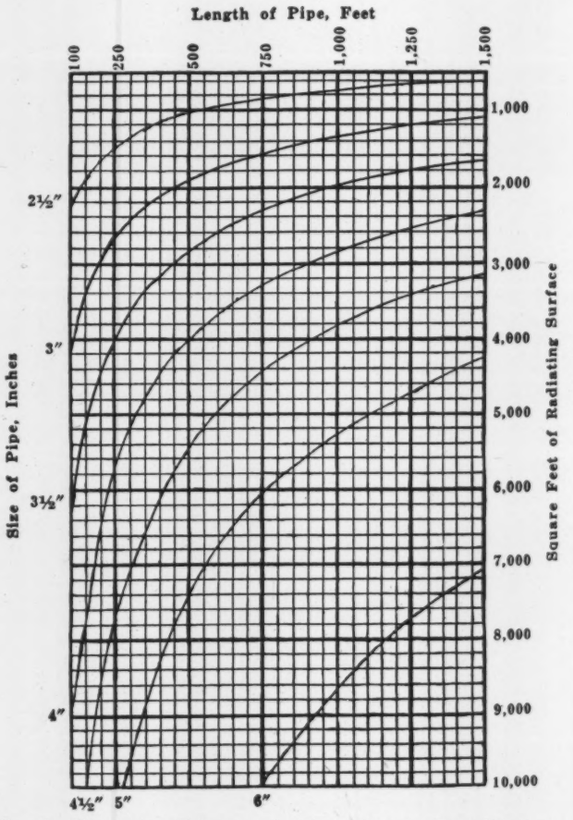


FIG. 1—CHART FOR DETERMINING PIPE SIZE TO GIVE DESIRED RADIATING AREA—1,000 TO 10,000 SQ. FT.

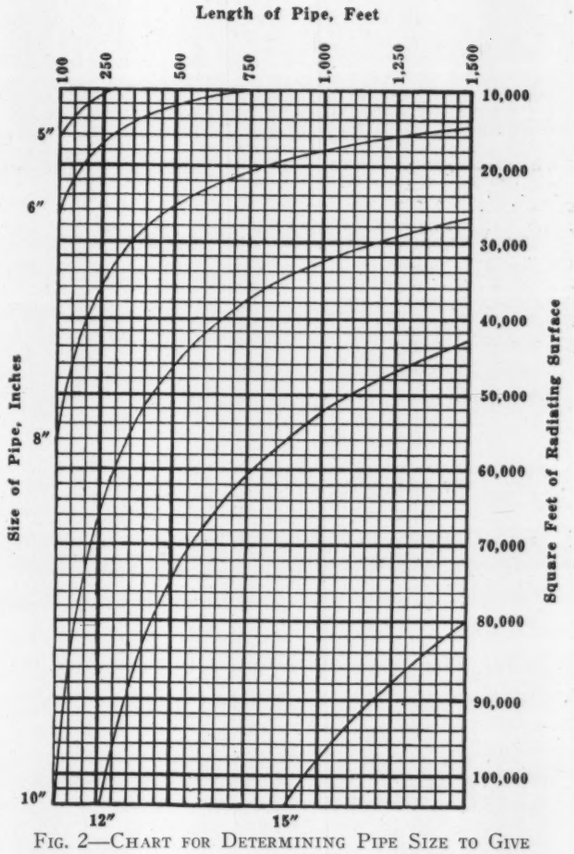


FIG. 2—CHART FOR DETERMINING PIPE SIZE TO GIVE DESIRED RADIATING AREA—10,000 TO 100,000 SQ. FT.

problem that would ordinarily, by other methods, require considerable figuring.

Another Example

Total radiating surface, 9,500 sq. ft. distributed from main as roughly sketched in Fig. 3.

Length of main, 500 feet.

Steam pressure supplied to radiators, 5 lb. gage.

Pressure loss through total length of main, 2 lb.

To solve this problem and problems of kindred type we recommend an orderly procedure along the following lines:

(1) From a line which indicates the main, sketch in the outlets as shown in Fig. 3.

(2) At each outlet indicate the sq. ft. of surface the outlet supplies.

(3) Multiply the factor, Table 1, for the steam pressure supplied the radiating surface by the factor, Table 2, for the pressure loss, or, .928 × 1.41 = 1.30 nearly. This factor 1.30 is used in paragraphs 5, 6, 7, 8, 9, and 10 below.

(4) Now examine Table 3 below. This table gives the sq. ft. of radiating surface supplied by pipes from ¾ in. to 2 in. diameter on the basis of 1 lb. steam pressure supplied to the radiators, and one lb. loss in pressure per 1000 ft. length of pipe.

Table 3

Diam. of pipe	¾-in.	1-in.	1¼-in.	1½-in.	2-in.
Sq. Ft. of Radiation	32	64	141	220	445

(5) This table will assist in approximating the pipe size. It shows that the last outlet will very likely be a 2 in. pipe. Multiply the surface, 445 sq. ft. for 2 in. pipe, by the factor Table 1, under the steam pressure, or, 445 × .928 = 412.9 sq. ft. Therefore a 2 in. pipe will supply the last outlet.

(6) Find the sq. ft. of surface a 2½ in. pipe will supply where the total length of the main is 500 ft., Fig. 1, and multiply by 1.30, or 1000 × 1.30 = 1300 sq. ft. Therefore a 2½ in. pipe will supply the last three outlets.

(7) Find the sq. ft. of surface a 3 in. pipe will supply where the total length of main is 500 ft., Fig. 1, and multiply by 1.30, or, 1900 × 1.30 = 2470 sq. ft. Therefore a 3 in. pipe will supply the last four outlets.

(8) Find the sq. ft. of surface a 3½ in. pipe will supply where the total length of main is 500 ft., Fig. 1, and multiply by 1.30, or, 2800 × 1.30 = 3640 sq. ft. Therefore, a 3½ in. pipe will supply the last six outlets.

(9) Find the sq. ft. of surface a 4 in. pipe will supply where the total length of main is 500 ft., Fig. 1, multiply by 1.30, or, 4000 × 1.30 = 5200 sq. ft. Therefore, a 4 in. pipe will supply the last nine outlets.

(10) Find the sq. ft. of surface a 4½ in. pipe will supply where the total length of main is 500 ft., Fig. 1, and multiply by 1.30, or, 5400 × 1.30 = 7020 sq. ft. Therefore, a 4½ in. pipe will supply the last twelve outlets.

(11) Find the sq. ft. of surface a 5 in. pipe will supply where the total length of main is 500 ft., Fig. 1, and multiply by 1.30, or, 7400 × 1.30 = 9620 sq. ft. Therefore, a 5 in. pipe will supply all of the outlets.

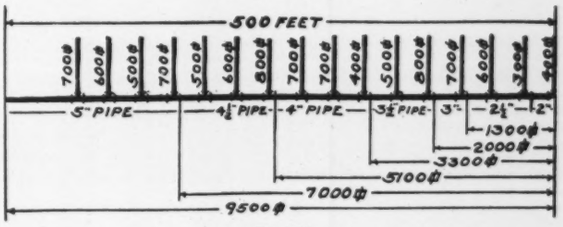


FIG. 3

When calculating the length of the steam main, allowance should be made for each elbow and globe valve as given in Table 4 below:

Table 4

Size	Allowance in Ft. for each elbow or globe valve						
	¾-in.	1-in.	1¼-in.	1½-in.	2-in.	2½-in.	3-in.
90° Elbow	2.75	3.50	4.60	5.35	6.90	8.25	10.25
Globe Valve	4.10	5.25	6.90	8.05	10.35	12.35	15.40
Size	4-in.	5-in.	6-in.	8-in.	10-in.	12-in.	15-in.
90° Elbow	13.40	16.75	20.20	26.60	33.40	40.00	50.75
Globe Valve	20.10	25.15	30.30	39.90	50.10	60.00	76.10

The members in the body of the table give the length in feet to be added for each elbow or globe valve on account of the resistance to the flow of steam.

Air Conditioning the Royal York Hotel

By B. E. Armstrong

THE Royal York Hotel, at Toronto, owned and operated by the Canadian Pacific Railways, was opened to the public on June 11, 1929. This huge structure, towering approximately 400 feet above the street level, consists of a solid block, 340 feet wide by 194 feet deep, and is the largest hotel and also the tallest building in the British Empire.

For the benefit of the reader who is not acquainted with the general construction of the hotel, a brief outline will facilitate his understanding of the various locations of the ventilation equipment and the service rendered by each unit.

Beginning at the lowest floor is the sub-basement containing the power plant, printing press, tailor shop, work shops, store rooms and fan rooms. The lower or basement floor, half a story below the sidewalk level, is reached by wide entrances from three streets and contains an arcade, exhibition hall, drug store, shops, bank, stock exchange, barber shop, coffee shop and grill room. Leading from this floor is a well-ventilated subway connecting the hotel with the train level of the Union Station. On the main floor is situated the lobby, lounge, main dining room, Venetian cafe, writing room, hotel offices, ticket office and main kitchen. Next comes the main mezzanine floor with its general offices, hospital, children's play room, private dining rooms of various sizes, library, writing room, lounge, ladies' hairdressing parlors and the lost and found office. On the convention floor, especially designed and set aside for convention purposes, is the concert hall, with stage, orchestra pit and the largest organ in Canada, banquet hall and ball room, with a combined area of 24,679 sq. ft., convention seating capacity for 4,070 persons, and banquet seating for 2,800 persons. Separate kitchen and service rooms are located in the rear of this floor. Then comes the convention mezzanine with a radio room, ladies' and men's retiring rooms, check rooms and the entrances to the galleries of the convention, banquet and concert halls. Next above the convention mezzanine is the first bedroom floor, of which there are seventeen. Above this is the roof garden where the supper dances are held. Adjoining the roof garden are the kitchens, foyer, men's retiring room and the check room. Overlooking the roof garden is the roof garden mezzanine, and rising above this again are the four elevator tower floors, the plenum chamber, and, at the very top, the architectural tower.

THE TALLEST BUILDING IN THE BRITISH EMPIRE

The Ventilating System

Prominent among the engineering features of the Royal York Hotel is the ventilating system, divided into five zones for simplification. This system includes 19 exhaust fans, generally located on the thirteenth bedroom floor, the first and second elevator towers, and principally the third elevator tower, twenty-six stories above the street. The exhaust fans located on the thirteenth bedroom floor exhaust through the roof at the set-back fourteenth bedroom floor, but without any interference with the bedrooms on that floor or with those above it. The fans in the elevator towers exhaust through the plenum chamber located above the fourth elevator tower floor. The air is discharged to the open through copper louvres. The exhaust fan for all the kitchens is independent of the above fans and has a separate exhaust discharge. In the bedrooms and public spaces, like amounts of air are entering and leaving in a given time. In the kitchens, however, this is not the case, the volume of exhaust air being considerably larger than the volume entering through the supply vents, so that a steady stream flows toward the kitchens from surrounding rooms, eliminating possibility of odors escaping into public quarters.

The air supply system comprises eleven fans, seven of which are located in the sub-basement in a large fan room, and four in the two fan rooms placed on the convention mezzanine floor. Approximately 300,000 cu. ft. of air per min. is supplied by the fans.

The seven fans in the sub-basement supply fresh air to the engine room, laundry, main dining room,

general basement, washrooms, main lobby and the lounges. The four fans located on the convention mezzanine floor supply the ball room, banquet hall, convention hall and the two sections of foyer and lounges.

The nineteen exhaust fans in the upper part of the building take care of the radio control room, general toilets, banquet hall, convention hall, ball room, roof garden and kitchen, engine room, laundry, general basement, main dining room, main lobby, kitchens, grill and coffee shop, and the west, central and eastern sections of bathrooms are served by three separate exhaust fans. These fans expel about 400,000 cu. ft. of air per min.

The air for the seven supply fans located in the sub-basement goes through a process of pre-heating, washing and re-heating. The air for the four supply fans located on the convention mezzanine floor is filtered. The air is washed or oil-filtered as required.

No refrigeration is used in connection with the ventilation system, nor is ozone or any similar treatment employed. The horsepower of the different motors employed varies from three to sixteen horsepower and totals approximately 200 horsepower, and the revolutions per minute average two hundred and ten. Direct current at 230 volts is used throughout the power system.

The fan casings, ducts, etc., were made of galvanized iron and painted where necessary. Some of the ducts are twelve feet across and extend from basement to tower. All the heaters connected with the ventilating system are thermostatically controlled, while the supply and the exhaust fans are all manually controlled.

Supply and Exhaust Fans

The following table will give the reader some idea of the size, capacity, etc., of the various supply and exhaust fans and what particular requirement each serves.

SUPPLY FANS

Place Supplied	*Cubic Feet of Air per minute*	*Minimum H. P.*	*R.P.M.*
Engine room	30,100	7½	178
Laundry	30,100	10	188
Main dining room	17,400	5	220
Private dining room, cafe and grill	23,900	7½	220
Lobby and lounge	22,000	7½	236
Basement, service rooms, private dining room service, main kitchen	45,400	15	190
Basement, sub-basement, miscellaneous rooms on main floor, miscellaneous rooms on main mezzanine	45,400	16	190
Banquet hall	30,100	8½	187
Convention hall	27,500	7½	184
Ball room	12,950	3½	250
Convention and banquet foyers, banquet hall, kitchen, etc.	31,400	9	190

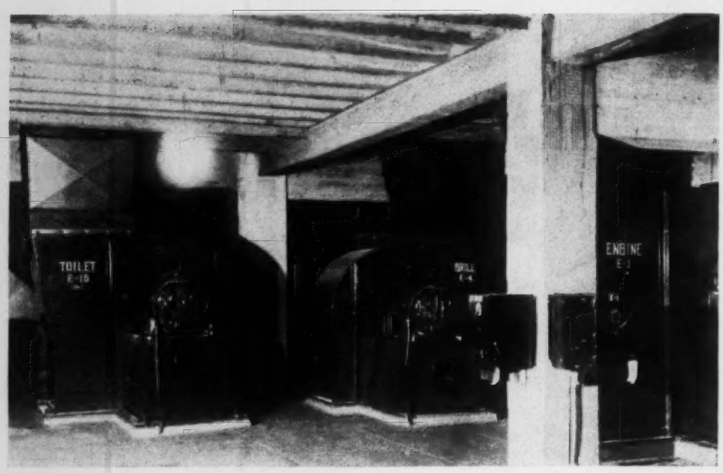
EXHAUST FANS

Place Ventilated	*Cubic Feet of Air per minute*	*Minimum H. P.*	*R.P.M.*
Engine room	30,100	7½	178
Laundry	34,100	9½	186
Main dining room	16,230	3	199
Private dining rooms, grill, cafe, lobby, lounge, grill room service, etc.	24,000	6½	224
Lobby, lounge, etc.	23,900	6	200
Basement service	27,500	7½	185
General sub-basement, miscellaneous rooms in basement, miscellaneous rooms on main mezzanine floor	35,600	10	169
Banquet hall	24,900	5	155
Convention hall	22,850	5	182
Ball room	10,220	2	205
Convention mezzanine and foyers	24,950	7½	203
Banquet hall kitchen	24,950	7½	203
Roof garden and mezzanine and kitchen	15,250	3	199
Bathrooms (east wing)	12,950	3	215
Bathrooms (center)	24,950	7½	203
Main kitchen	48,700	15	150
Bathrooms (west wing) barber shop	18,500	6	242
Bathrooms (west wing)	12,950	3	215

Owing to the fact that this system has only been in operation during the last three summer months, it is impossible to give the actual results of air conditioning in the colder weather. However, during the summer months superior air conditions have been maintained. During the hottest weather experienced in Toronto, the temperature in the Royal York Hotel remained at an agreeably low point. Provisions have been made in connection with the ventilating system to maintain an average temperature of 70 deg. fahr. at an outside temperature of 20 deg. below zero.

It is found to be preferable to keep the windows closed in hot weather on the belief that an open

THE VENETIAN CAFE ON THE MAIN FLOOR OF THE ROYAL YORK



Exhaust Fans in the Elevator Tower

window interferes more or less with an efficient circulation of air. Therefore, the effect of closed windows on a hot day would be a perfect circulation of cool air throughout all the rooms.

The Heating Equipment

Steam for the Royal York will be produced in a central power plant, which will supply also the new Union Station and the C. P. R. and C. N. R. express building. The pipe lines are brought to the hotel from the power plant through a waterproofed, concrete lined tunnel. The steam enters the hotel engine room at a pressure of 150 to 180 lb., through two pipes, one 8 in. and the other 10 in. in diameter, the smaller pipe being designed to take care of summer requirements and the larger one the winter demand. The pipe not in use acts as a spare, and on occasions of very heavy overloads both supply lines can be utilized at the same time. Two electric flow meters keep an accurate check of the amount of steam entering the hotel. The steam is brought through the engine room on the loop system and is reduced to the following pressures:—100 lb. for use in the laundry; 40 lb. for use in the kitchens, of which there are five; and 1 and 2 lb. for heating purposes.

Installed throughout the bedrooms is a new system of heat control. The building has been divided into three separate zones. Each zone is equipped with its own apparatus (three thermostats, three air compressors and three control sets) which is specially designed to conform to the requirements of the separate zones. Controlling equipment for the entire system is located in the chief engineer's office, where, by the movement of a lever, the amount of steam supplied can be increased or decreased at will.

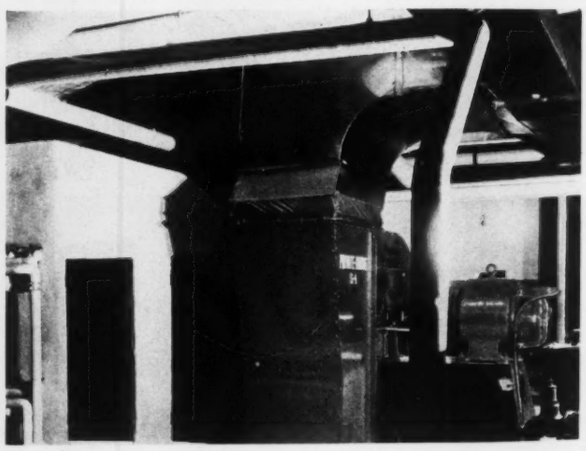
The three zones are basement to convention mezzanine floor, first bedroom floor to fourteenth bedroom floor, and the fourteenth floor to the roof. Exhaust steam flows through an 18-in. main from the basement to the top of the building, a distance of approximately four hundred feet. From this, branches are taken off throughout the building. To take care of condensation there are two double-suction centrifugal pumps.

More than thirty-six miles of asbestos pipe insulation has been applied on the high, intermediate and low pressure steam lines and on all hot water, cold water and return lines in order to provide the last word in modern comfort and efficiency to the guests of the hotel. From the time the steam is generated at the Toronto Terminals heating plant at the foot of York Street, it is controlled by asbestos insulating materials.

Not only are all pipes adequately insulated, but all heater casings, smoke flues, cold air intakes, condensation tanks, water softening apparatus and all the multitude of various types of equipment as well. More than five carloads of special asbestos cement were used. 26,000 yards of heavy canvas were applied to the finished insulated surfaces. The type of insulation used on the various piping follows:—High-pressure steam piping, sponge felt; low-pressure steam pipes and hot water piping, 85 per cent magnesia; cold water piping, anti-sweat; hot water heaters, magnesia and two coats of cement; water softeners, two layers of 1-in. hair felt, finished with two coats of cement.

Refrigeration

The refrigerating equipment consists of three 36 ton vertical enclosed-type ammonia compressors, each of which is direct connected by special flexible couplings to a horizontal poppet valve steam engine.



Private Dining Room Supply Fan Showing Starting Motor in Foreground and Fan Connected with Air Washer, Pre-Heater and Re-Heater

There are two 400 gallon per minute, direct connected electrical driven, centrifugal brine pumps, which circulate brine through the refrigerators in the various kitchens throughout the building; two 150 gallon per minute, direct connected, electrical driven, centrifugal brine pumps which circulate brine at zero temperature through the ice cream freezers and sharp freezers for the storage of sea food, game, etc.; two large ice-making tanks for the manufacture of clear crystal ice; and also a four ton, electrical driven refrigeration plant, thermostatically controlled, on the second tower floor, which circulates

cold brine through the various refrigerators in connection with the roof garden kitchen.

Engineers and Contractors

The ventilation system was installed by James Ballantyne, contractors, of Montreal, and designed by Walter J. Armstrong, engineer of the firm of Ross and Macdonald, architects, of Montreal, and Sproatt and Rolph, associate architects, of Toronto. The general contractors and construction engineers were Anglin-Norcross Ltd. of Montreal and Toronto.

The various details and problems to be met during construction were handled by the members of the staffs of the above, together with the department heads and the engineers of the Canadian Pacific Railway.

Heating and Ventilating a Newspaper Plant in Toronto

THE new home of the Toronto Star, a twenty-three story skyscraper with three levels of basements, is heated with a vacuum system by which the required temperatures can be maintained using a wide range of vacuums. Steam is supplied by two 225 hp. water tube boilers; a 60 hp. water tube boiler is installed for use during the summer months.

THE TORONTO STAR BUILDING, IN THE MODERN ARCHITECTURAL STYLE

Ventilation of offices is by means of windows hinged in the middle and opening outwards their full length; when partly open, a current of air is created at the top and at the bottom as well as at the sides. All of the offices have outside windows.

The composing room is ventilated by side windows fitted with window type exhausting fans in addition to saw-tooth skylights, which are controlled with hand-operated opening gear re-line shafts, mitre gears and turning handles. A fan of 20,000 cubic feet per minute capacity supplies the main press-room, reel room, and daily paper storage room with water-cooled air. The air is introduced to the rooms through under-floor duct systems and floor grates; the air is exhausted at various heights and is returned through sheet iron ducts to the air washing and conditioning chamber. Temperature and humidity is kept constant. Unit heaters supply air, which is exhausted with fans fitting in the outside windows, for the mailing room.

The substation in the basement, receiving current at 2,200 volts from Niagara, is ventilated by means of a horizontal, direct-connected fan with a capacity of 3,500 cubic feet per minute. The air is brought in through air filters and ducts at the street level and an exhaust fan of similar capacity discharges the used air through a shaft to a point at the second floor level.

The Toronto Star building contains thirty miles of pipe of many kinds, from steam pipes and message tubes to ink and liquid soap pipes.

Third National Fuel Meeting

The third national fuels meeting of the American Society of Mechanical Engineers will open October 7 at the Bellevue-Stratford hotel in Philadelphia. The meeting will continue through October 10 and is expected to be one of the most outstanding programs ever staged by the fuels division. An effort has been made to include the recommendations of other organizations in planning the program set-up, and the conference will take the form of a fuels forum where all engineers may come for aid and exchange of experiences. The major divisions of the program include papers on subjects of general interest, power plant problems, low-temperature carbonization, industrial fuel problems, domestic heating situation, stokers, and smoke abatement.

Air Conditioning in Telephone Exchanges

By Malcolm C. W. Tomlinson

THE telephone business has grown to amazing proportions within the past fifty years. Yet comparatively few people have any concept of the extent of its growth or of the engineering feats which have been necessary to make the telephone and the service which goes along with the telephone what they are today.

The telephone business may be considered as formed by three main divisions. First of all is the preparative engineering work, then the manufacturing and, last but not least, the service. It is the heart of the service end of this business with which we are concerned when we discuss various phases of telephone exchange engineering. Here all connections are made for both local and long distance calls. Here the harassed operator makes the manual connections and here the intricate switching apparatus functions without the operator's aid.

As far as a visitor can see, the exchange consists of a substantial building which houses a large amount of cable, wires, intricate metal parts mounted on metal racks or frames and switchboards made with a mahogany casing. Young girls with nimble fingers control this mysterious equipment.

Actually, the situation is somewhat different. The communicating system with the outside world consists of lead covered cable, loading coils and repeating coils. This cable is made up of copper wires and lead sheath. Each wire is wrapped with a paper insulation and is twisted with adjacent wires to form pairs or quads (two pairs of wires). These wires vary in size from 13 to 26 (American wire gage) depending on the type of service and they usually number from 52 to 2424 in a single cable of the lead covered type. The pairs and quads of wires are stranded into a cable core in such a manner as to form definite layers. The paper insulation is colored for identification purposes so that each layer of wires may be readily recognized. This cable core, as a protection against weather and rough usage, is covered with the lead sheath previously mentioned. The repeating and loading coils resemble small and large transformers, respectively, and are constructed for the purpose of amplifying sound. The connections between the switchboards and the lead covered cable are made with copper wires or cables.

The latter, or "factory", cable consists of wires separately covered with a cotton insulation and bundled together, without any twist or lay, and wrapped at half inch intervals with a stout cord. This cable is usually coated with beeswax or paraffin as a protection against moisture.

Electrical Leakage Due to Dust and Moisture

Telephone communication is carried on by means of very low alternating currents. For this reason the same problems arise which obtain in laboratories or on test floors where delicate electrical measurements must be made. One of these problems is electrical leakage due to dust or to fine films of moisture. This leakage may occur through the insulation or across the surfaces of electrical equipment.

The other problem is to secure moisture proofed insulation. In the latter case very much better electrical characteristics are to be had with equipment so protected. Not only can the insulation resistance be greatly increased and the conductance bettered by such insulation but the capacity, which depends to a certain extent on the other two electrical factors just named, also is improved.

Air Conditioning the Answer

At first glance these problems seem quite simple and, for the laboratory, they are. But where enormous quantities of equipment must be installed constantly to meet maintenance requirements and service demands, first cost becomes a considerable factor. Thus, air conditioning has an opportunity in the telephone exchange for it not only can reduce the dust problem to a minimum but will also provide suitable protection against moisture regain on the part of insulated electrical apparatus. Furthermore, where such apparatus has given trouble due to moisture already adsorbed from the atmosphere, the introduction of a low relative humidity, constantly maintained, will quickly remedy such troubles as can readily be seen from Fig. 1. There, moisture regain and loss for cotton is plotted against per cent relative humidity. It will thus be apparent to the reader that, as all hygroscopic materials have similar regain curves and as most electrical insulation is hygroscopic in character, a regain due to high relative humidity is sure to be followed by a loss of moisture as the relative humidity is lowered and also that it is possible to engineer such problems with considerable accuracy as far as moisture removal is concerned.

Factory Cable the Problem

Naturally, providing it has been properly dried and also providing the lead sheath is not cracked, lead covered cable in the telephone exchange presents no problem, since it is usually well dried, except at the ends where the lead sheath and lead cap must be removed for splicing. Factory cable, never dried but protected with beeswax or paraffin, is probably the weakest point in telephone exchanges as far as moisture is concerned. It is made up in varying forms to suit each particular job and is not, at the present time, dried in the factory.

Dependent on the prevailing atmospheric conditions at the time of manufacture, factory cable insulation naturally contains more or less moisture. As beeswax and paraffin are crystalline in structure and as such coatings

are easily cracked in handling the cable there is no reason to believe that moisture can be kept out of the cotton insulation. The insulated cords and wires are never dried and seldom coated to prevent regain of moisture.

Repeating and loading coils have also been a source of much trouble. Recently steps have been taken by manufacturers to subject these coils to a vacuum drying process, followed by impregnation under vacuum, and a constantly maintained relative humidity not to exceed twenty per cent and, in most cases, as low as ten per cent. It is even probable that portions of large exchanges, equipped with apparatus that has been poorly dried, will see the way to make use of relative humidities as low as one-half to five per cent.

Still a Matter of Research

No attempt has been made in this paper to go beyond considerable improvement has been secured as the product is thus freed of much moisture and as regain of moisture is prevented. As far as the automatic and machine switching equipment of telephone exchanges is concerned it is apparent that semi-automatic and automatic electrical equipment will give more trouble, due to leakage, than manually operated equipment as the number of metal parts and contacts are largely increased.

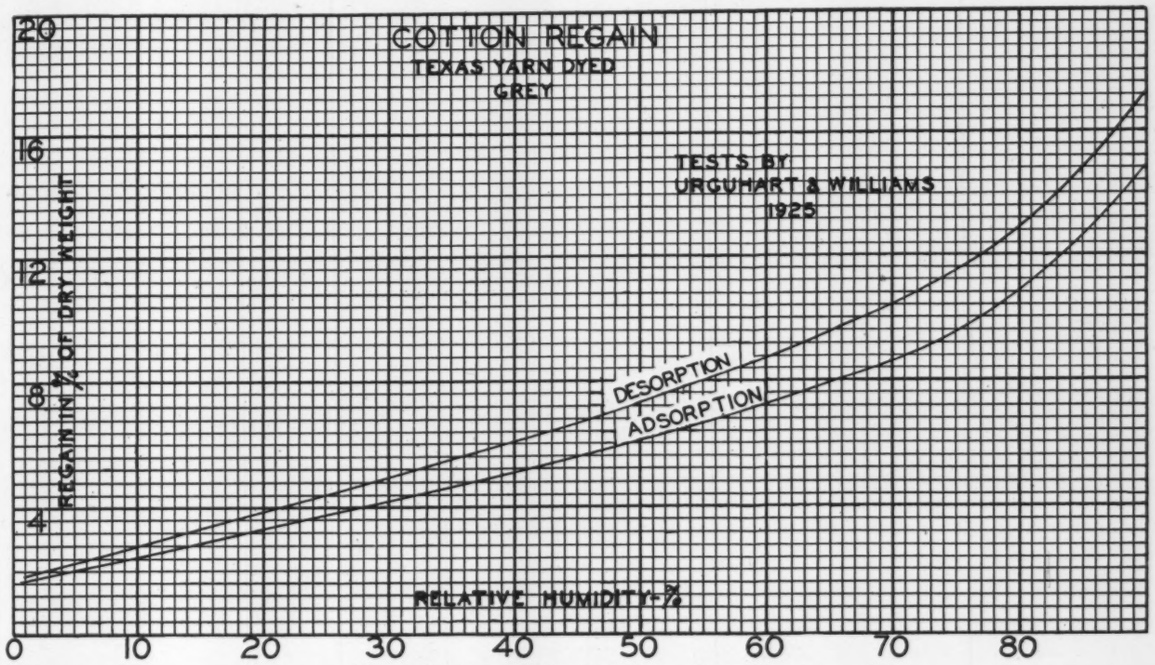


FIG. 1. MOISTURE REGAIN AND LOSS FOR COTTON

Air Conditioned Production Will Help

The preceding discussion gives a fairly clear picture of the difficulties in telephone exchanges which can be overcome largely by air conditioning. In passing it should be borne in mind that these difficulties will be eliminated to a considerable extent when telephone equipment can be produced, at the source of supply, which has been thoroughly dried and sealed against regain of moisture.

Air conditioning in the telephone exchange is practically ten years old. It has been practiced mainly in localities subject to high humidities, such as the territory adjacent to the Virginia capes, and at first consisted of fan equipment solely. The cost of air conditioning at that time was considered excessive, as the only result expected was the elimination of humidities above fifty or sixty per cent. Later forty per cent was considered good practice. Today, with the results of drying and air conditioning investigations covering the past six years, the telephone exchange is faced with the drying problem for equipment already installed. This means plain statements of facts except to predict future trends. The production of very low relative humidities is an infant branch of air conditioning where research work is still needed. Furthermore, human comfort under such conditions requires temperatures much above the usual 65-70 deg. fahr. range. It is to be expected, therefore, that advancement along this line will be somewhat slow for a few years.

A. S. R. E. Anniversary

The twenty-fifth anniversary of the *American Society of Refrigerating Engineers,* to be celebrated in New York city the first week in December, will be a high spot in the history of refrigeration, attracting the full attention of its various fields. This was the opinion expressed at the recent council meeting at the Society headquarters, when the program committee outlined plans for this event.

The meeting will last four days, December 4, 5, 6 and 7, held at the Pennsylvania Hotel and the Engineering Societies Building, 29 West 39th Street. Each session will be devoted mainly to some specific subject such as ice plants, domestic refrigeration, cold storage, etc. There will be a joint session with the *American Society of Mechanical Engineers.*

At each of the technical sessions, a prominent industrialist will make the keynote address. This is a new arrangement and is intended to shed light on the present status of refrigeration.

Fuel Oil Pumping and Piping

By H. L. Kauffman

THE GROWING use of oil as fuel in industrial plants is causing numerous plant construction and maintenance engineers and executives to want to know more about the layout and design of fuel oil pumping and piping systems, and how best to keep that system functioning effectively after it has been installed. Consequently, the purpose of this article is to present such information about the handling of fuel oil in industrial plants as will aid the reader in designing a new system for maximum effectiveness, or to alter an existing system so that maintenance and operating troubles will be reduced to a minimum.

Fuel oil is used as a source of power for generating steam in all industries; but it is also widely employed in forge-shop practice, in boiler manufacturing and repair shops, in steel-foundry practice, in malleable-iron foundry practice, as fuel for copper matting and refining furnaces, for heat-treating furnaces, and, merely to mention a few, as fuel in such specific industries as the glass industry and the sugar industry, and in the operation of such specific equipment as lime kilns, and cement kilns and dryers.

Each type of equipment or plant must be taken into consideration when designing a particular fuel oil pumping and piping system, as the operation of a specific unit or plant may necessitate changes in design different from that of any other type of unit or plant. However, the effort will be made so to discuss the subject herein as to be applicable to all industrial plants.

Grade of Fuel Oil to Be Handled Must Be Known in Designing System

In planning an installation for handling fuel oil, one must first have information as to the physical characteristics of the oil to be pumped through the line. This is necessary so that one will know whether or not to provide for heating equipment or steam lines for keeping the oil sufficiently fluid so that it can be pumped readily. Bear in mind that some fuel oils with a high paraffin content will flow easily at moderate temperatures, but when these same oils are subjected to cold-weather conditions they become difficult to pump. Hence, the approximate cold test of the fuel oil to be used should be known, as well as the density and viscosity of the oil at various temperatures within range of the oil temperatures attained during the transportation of the oil through the line.

As an aid in deciding what grade of fuel oil will most nearly fit the needs of a particular plant, Table 1,

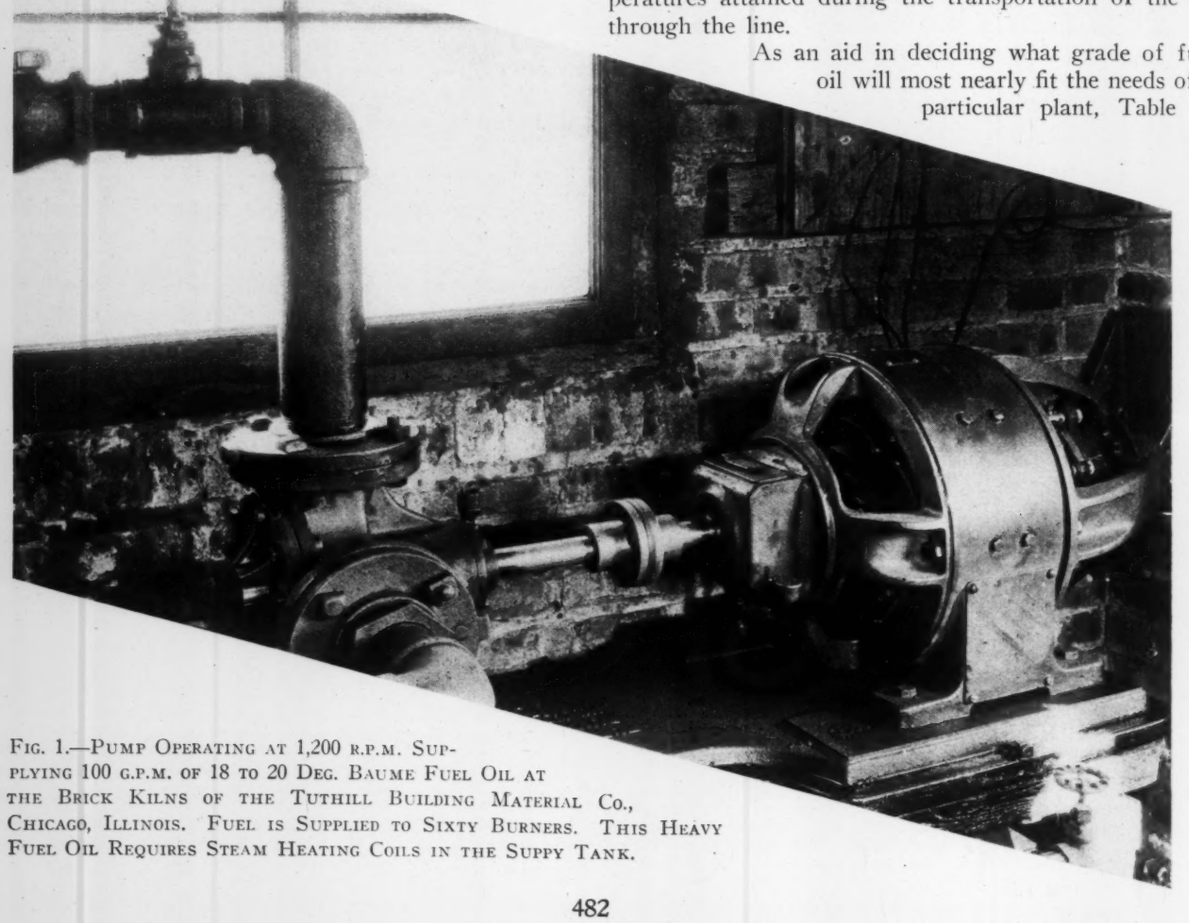


FIG. 1.—PUMP OPERATING AT 1,200 R.P.M. SUPPLYING 100 G.P.M. OF 18 TO 20 DEG. BAUME FUEL OIL AT THE BRICK KILNS OF THE TUTHILL BUILDING MATERIAL CO., CHICAGO, ILLINOIS. FUEL IS SUPPLIED TO SIXTY BURNERS. THIS HEAVY FUEL OIL REQUIRES STEAM HEATING COILS IN THE SUPPY TANK.

A discussion applicable to any industrial plant using fuel oil

showing the specifications for various grades of fuel oil as adopted by the American Oil Burner Association, is given.

Further commenting upon the effect of the grade of oil used on the design of the system, it may be said that in general an oil below about 20 deg. A.P.I. in gravity should be heated in order to reduce its viscosity so that it can be pumped without difficulty. In most cases, it is advisable to heat the oil to 30 deg. fahr. below its vaporizing point and, after it leaves the pump, to pass said oil through an oil heater, the function of which is to heat the oil to just below its vaporizing point in order to secure from it the highest efficiency and strictest economy.

General Types of Fuel-Oil Pumping Systems

Before giving any specific hints on the selection and installation of pumps, pipe and fittings of a fuel-oil system, it may be well to mention the general types of such systems now in use. Figure 3 shows an imperfect form of fuel oil-pumping system. In studying this drawing, note especially the following points:

1. But one pump is used, located as shown. No reserve pump is provided.
2. There are no oil heaters or pump regulators.
3. The suction pipe is higher than the pump and is taken out of the top of the tank.
4. No combination foot valve and strainer is used with this system.
5. After the oil passes from the pump the supply pipe is raised, causing vapor pockets.
6. The oil does not circulate and the pipes are so laid as to cause a "dead end" upon the pipes.
7. The relief valve is located in the manner shown and the overflow pipe is coupled to the suction pipe of the pump.

Every part of the assemblage of this oil system is imperfect.

Figure 4 shows a modern and perfect form of a fuel-oil handling system. In this illustration, the attention of the reader is especially directed to these installation features:

1. Two pumps are used—one for reserve. Each pump is of adequate capacity to supply oil to the two boilers.
2. The pumps are placed upon drip pans and stands, and under each pump is located a modern oil heater provided with thermometers, etc.
3. Each pump is provided with a pump regulator so that if the pressure relief valve does not function, the steam operating the pump will cause the pump to stop by means of this regulator.
4. Pulsometers are provided, their function being to remove pulsations of the pump.
5. A combination foot valve and strainer is provided in the tank.
6. A heater is provided in the tank to heat the oil to reduce its viscosity,

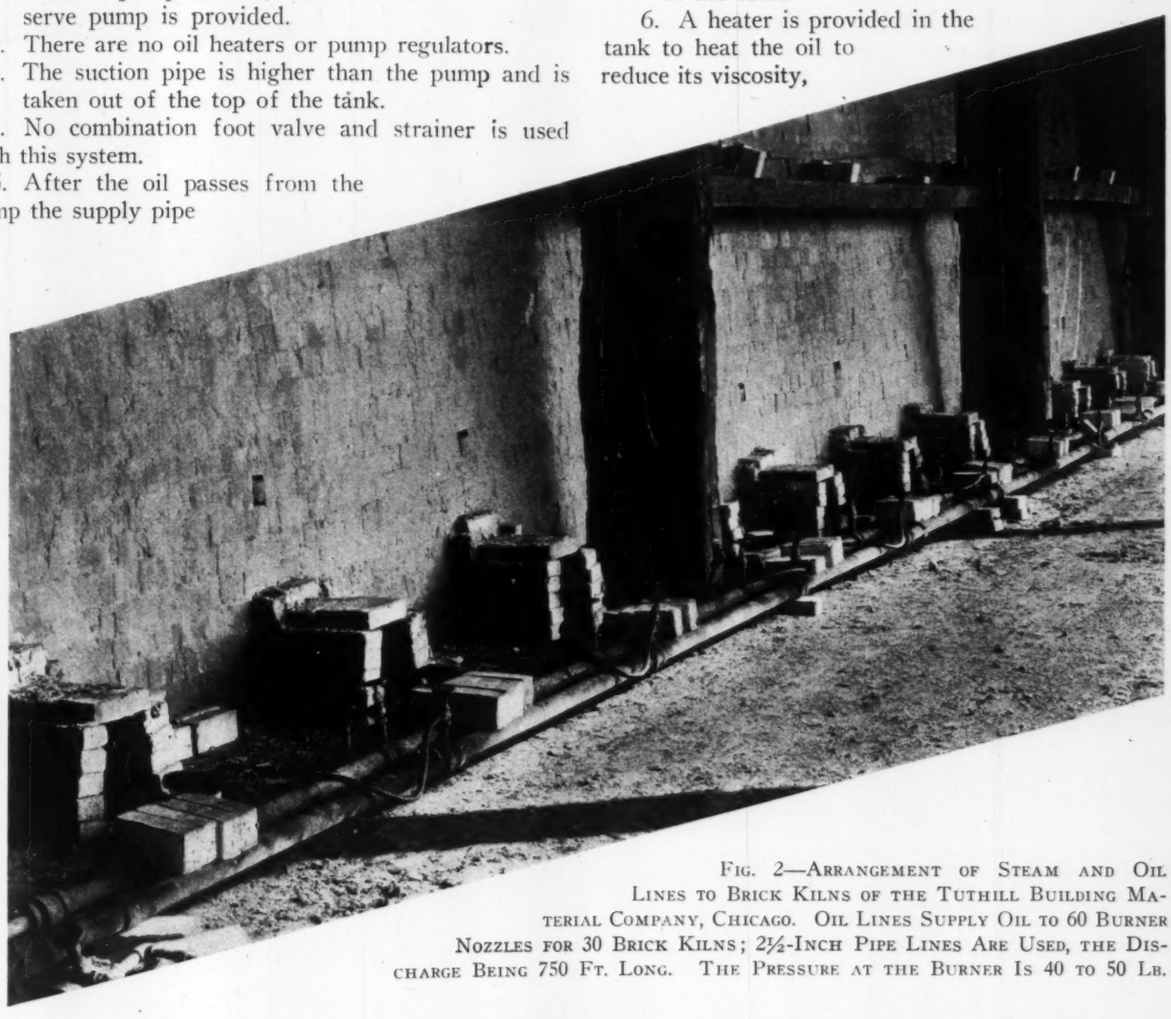


FIG. 2—ARRANGEMENT OF STEAM AND OIL LINES TO BRICK KILNS OF THE TUTHILL BUILDING MATERIAL COMPANY, CHICAGO. OIL LINES SUPPLY OIL TO 60 BURNER NOZZLES FOR 30 BRICK KILNS; 2½-INCH PIPE LINES ARE USED, THE DISCHARGE BEING 750 FT. LONG. THE PRESSURE AT THE BURNER IS 40 TO 50 LB.

if the oil is below about 20 deg. A.P.I. gravity.

7. The oil supply pipe circulates the oil so that there are absolutely no "dead ends," and if the oil must be heated, it can be delivered to all the boilers at the temperature required.
8. The relief valve is located at the last boiler in the battery and the overflow pipe from the relief valve passes back into the tank independently of any other pipe line, and does not couple to the suction pipe of the pump.
9. The suction pipe, the oil supply pipe to the burners, and the overflow pipe are provided with a steam pipe running parallel with and immediately under these pipes in order to reduce the viscosity of the oil remaining in these pipes and to keep it in a fluid state. This is very necessary, especially when the plant is shut down over Sunday, for the oil in the suction, supply and overflow pipes will acquire a jelly-like consistency if this steam heater pipe is not provided.

This system shows but two boilers in battery, but as many as forty-six may be included in one system. In fact, systems 6,000 feet in length have been used.

In still another general type of installation, modern in every respect, oil heaters are provided immediately under the pumps. In usual practice both pumps exhaust into one heater, while the other heater may be connected with live steam from the boiler in order to insure the heating of the oil to the proper temperature. Live steam, how-

Table 1

Specifications for Furnace and Fuel Oils

(Adopted by the American Oil Burner Association)

Name of Oil	Scope	Flash Point		Water and Sediment Maximum	Pour Point	Distillation Test	Viscosity Maximum
		Minimum Legal or Pensky Closed Tester	Maximum Pensky Martens Closed Tester				
No. 1 Furnace Oil—Light	A light distillate oil for use in burners where a high grade fuel is required	110° F.	165° F.	Trace	0° F.	10% reading, not over 420° F. End point not above 600° F.	
No. 2 Furnace Oil, Medium	A grade of oil for use in burners where a high grade fuel is required	125° F.	190° F.	Trace	0° F.	10% reading, not over 440° F. 90% reading not over 620° F.	
No. 3 Furnace Oil, Heavy	A light fuel oil intended for general industrial consumption and for use in burners where a medium grade fuel is required	150° F.	200° F.	0.1%	0° F.	10% reading, not over 460° F. 90% reading not over 675° F.	55 seconds (Saybolt at 100° F.)
No. 4 Fuel Oil, Light	A light fuel oil intended for general industrial consumption and for use in burners where a medium grade, medium viscosity fuel is required	150° F.	250° F.	1.0%			125 seconds (Saybolt Universal at 100° F.)
No. 5 Fuel Oil, Medium	Similar to a grade of fuel oil listed in the U. S. Government Specifications as Bunker Fuel Oil "B." It is suitable for burners adapted to low grade oil of medium viscosity	150° F.		1.0%			100 seconds (Saybolt Furol at 122° F.)
No. 6 Fuel Oil, Heavy	Similar to a grade of fuel listed in the U. S. Government Specifications as Bunker Fuel Oil "C." This oil is suitable for burners that are adapted to a low grade of fuel oil of high viscosity	150° F.		2.0% (Sediment Maximum 0.25%) A deduction in quantity will be made for all water plus sediment in excess of 1%			300 seconds (Saybolt Furol at 122° F.)

Note: The maximum pour point shall be 0 deg. F. except for localities where the average temperature between October 1st and May 1st is 40 deg. F. or above, in which case the pour point may be plus 15 deg. F.

The first four oils are required to be "a hydrocarbon oil free from water, acid, grit and fibrous or other foreign matter likely to clog or injure the burner or valves." The medium and heavy fuel oils, if required, "shall be strained by being drawn through filters or wire gauze of 16 meshes to the inch. The clearance through the strainer shall be at least twice the area of the suction pipe and the strainers shall be in duplicate."

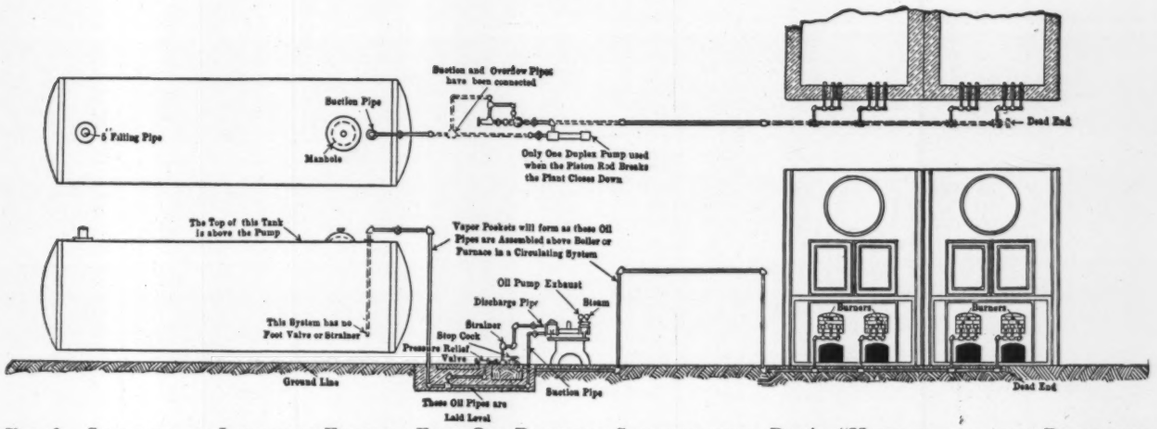

Suction and Overflow Pipes have been connected
Suction Pipe
6" Filling Pipe
Manhole
Dead End
Only One Duplex Pump used when the Piston Rod Breaks the Plant Closes Down
The Top of this Tank is above the Pump
Vapor Pockets will form as these Oil Pipes are Assembled above Boiler or Furnace in a Circulating System
Oil Pump Exhaust
Discharge Pipe
Steam
Strainer
This System has no Foot Valve or Strainer
Stop Cock
Pressure Relief Valve
Burners
Burners
Ground Line
Suction Pipe
These Oil Pipes are Laid Level
Dead End

FIG. 3.—SHOWING AN IMPERFECT FORM OF FUEL-OIL PUMPING SYSTEM; FROM DAY'S "HANDBOOK OF THE PETROLEUM INDUSTRY"

ever, is only used when heating heavy viscous oils.

Unloading Fuel Oil from Tank Car to Storage

Fuel oil is usually delivered to industrial plants in tank cars, though in plants of smaller size it is often bought from local jobbers and delivered by truck. In the latter case there is no problem involved in unloading, for it usually means nothing more than to let the oil flow by gravity to an underground oil-storage tank.

When bought in tank cars, however, special provision must be made for transferring the oil from the car to storage. Both under- and above-ground tanks are in use for storing fuel oil, but a discussion of their design and construction, and the advantages of each, is not considered as being within the scope of this article.

Figure 6 shows the approved method of unloading a tank car through the bottom outlet with a galvanized steel, flexible hose. A heavy 45-degree elbow tank-car reducer coupling is soldered on one end and a female swivel coupling on the other. Usually a 10-foot length of hose is sufficient. Flow is by gravity to an underground storage tank, if located close to the tank car; by pumping to an above-ground tank or if underground storage is very far from the tank.

Often, the use of overhead unloading apparatus exclusively is given preference. With such equipment, upright posts should be 8 feet from the nearest rail, while the platform must not extend more than 2 feet from the posts toward the track. Posts should be about 18½ feet long, 15 feet of which should be above ground. Suction pipe from the base of posts to the pump house should be raised an inch or so above the ground, if possible, since by so doing it is easier to locate any leaks.

Figure 7 gives the details and measurements of an approved combination overhead and underneath unloading apparatus for tank cars. Use of such apparatus is good practice, inasmuch as it is occasionally necessary to resort to either method of unloading. The swing

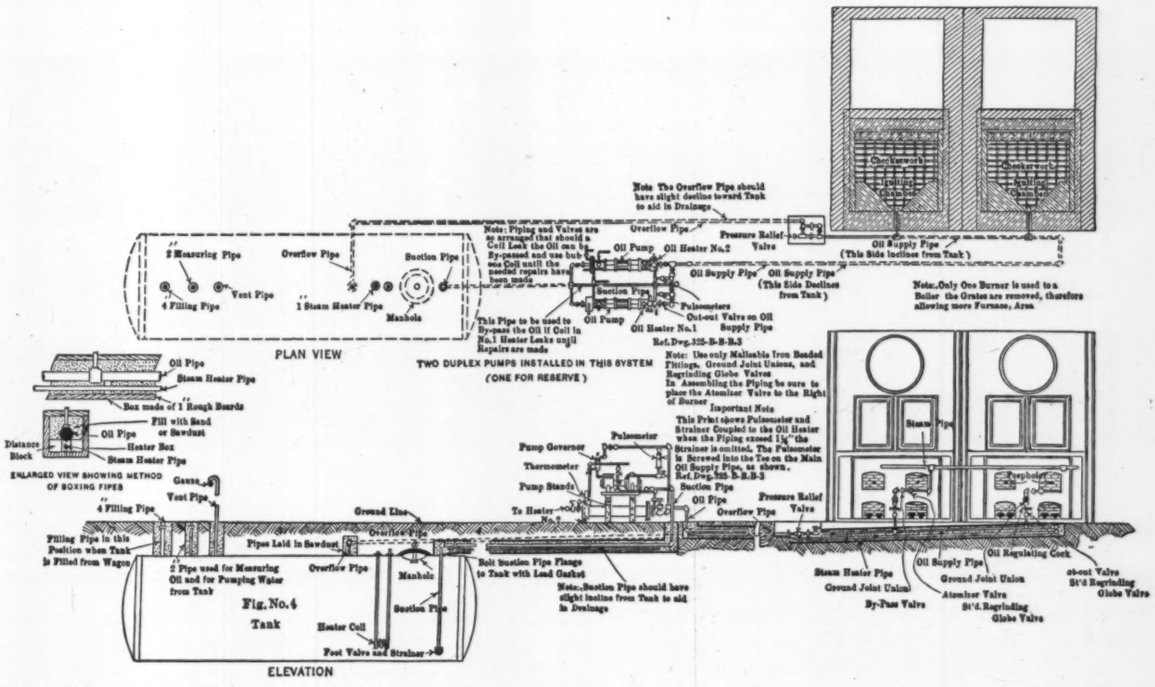


FIG. 4.—SHOWING A MODERN AND PERFECT FORM OF A FUEL-OIL HANDLING SYSTEM; FROM DAY'S "HANDBOOK OF THE PETROLEUM INDUSTRY"

FIG. 5.—FUEL-OIL PUMPS AND PIPING IN THE BOILER ROOM FOR AN APARTMENT BUILDING

arms are attached to uprights with a double swing joint. This provides easy access to the dome of the tank car. When not in use it is advisable to lock the swing arms to the uprights. Upon arrival of a car the swing arm is placed on the tank car platform, the suction pipe is inserted through the dome of the car and then connected to the swing arm with a union wrench coupling. A two-suction line rack is recommended if more than one grade of fuel-oil is to be unloaded. Uprights should be erected 10 feet 9 inches from the center of the railroad track, as shown in the illustration.

Especial attention is called to the fact that it is advisable to use a strainer connection in conjunction with any type of unloading apparatus to keep sediment and foreign matter from entering the storage tanks.

Another method often employed in unloading fuel oil by gravity, and one that is time-saving, consists in placing a 10-in. pipe in the ground between the rails with 10-in. openings, 43 ft. center to center, corresponding to the length of the longest tank car. These openings are 10-in. nipples, threaded to receive pipe caps when they are not in use. By such an arrangement, it is possible to unload several cars at the same time, the 10-in. pipe being large enough to receive the oil from the 4-in. discharge pipes of the car when its valve is wide open. In this way, with the oil in the tank car heated as may be necessary, a tank car can be emptied in 30 minutes, as compared with about 3 hours when unloading a tank car by means of a 3-in. pipe or flexible hose.

A very important fact to keep in mind is that the connection between the car and the discharge pipe should be capable of withstanding a pressure of 10-ft. head of oil; or, else, the receiving pipe should be large enough to carry the oil away without any connection whastoever between the pipe and the car.

Tank Connections

Figure 9 shows the openings on a typical horizontal underground fuel-oil storage tank.

Above-ground fuel-oil tanks, in addition to inlet and outlet connections, should be provided with flanges for water draw-off, with heating coil inlet and outlet (so that the more viscous oils can be readily pumped), and adequate roof vents. In addition, fire protection can be obtained by the use of either steam-smothering lines or "foam" lines.

For removing fuel oil from a tank, a "swing-suction" line should be used. Such a line, after passing through the tank shell, is connected inside the tank by means of a two-elbow swing to a length of pipe, the open end of which can be raised or lowered by means of a cable passing through the roof of the tank. By this arrangement it is possible to draw oil from any desired level, a matter of importance in case of stratification. The swing suction is also valuable when connections outside the tank are broken or valves become jammed since, in such instances, the pipe may be raised until it is above the liquid level preventing any flow of oil from the tank.

When fuel-oil tanks are fitted with steam-smothering lines, such lines should be brought through the top ring of the tank and should have a goose-neck higher than the roof angle, so that oil can not flow into the steam pipes when the tank is completely filled. The lowest part of the branch before entering the main should be equipped with a valve for draining condensate.

In order to receive the advantages gained by the use of the heavier, lower-priced fuel oils, storage tanks

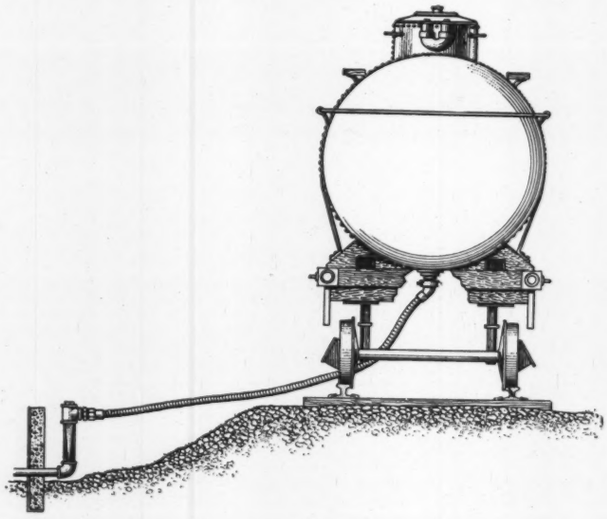


FIG. 6.—SHOWS THE APPROVED METHOD OF UNLOADING THE TANK CAR THROUGH THE BOTTOM OUTLET WITH A GALVANIZED STEEL, FLEXIBLE HOSE

should be provided with heating coils so that pumping of so viscous a product can be done at about 100 deg. fahr. For small, low tanks a flat coil over the entire bottom is all that is necessary. Such a coil should be supported on a wooden frame work about 6 in. from the bottom and so arranged that the piping will drain toward the outlet. If the coils were placed in direct contact with the bottom of the tank, much of the heat would be

transferred thereto and radiated without heating the oil at all.

For larger tanks, where the cost of heating the whole tank would be great, a coil around the suction pipe only generally will be sufficient. In the latter case, a box-type coil is most satisfactory.

For the vertical fuel-oil tanks, the vertical type of coil is to be preferred. Such nests of coils should be installed in sections and should be placed at least six inches from the shell of the tank, so that there will be sufficient area for the descending oil stream at this point—for, to secure efficiency in heating, adequate circulation of the oil in the tank is necessary and the desired circulation can not be obtained with the coils against the tank shell. Likewise, so that the total area of the pipe will be presented to the oil, there should be at least a 2-in. space between each turn of the coil.

Since rapid heating of oil in a tank can not take place when coils are half filled with air or water, piping should be so pitched that rapid drainage of the condensed steam will result. Waste of steam can be minimized by fitting the end of the outlet pipe with a suitable steam trap

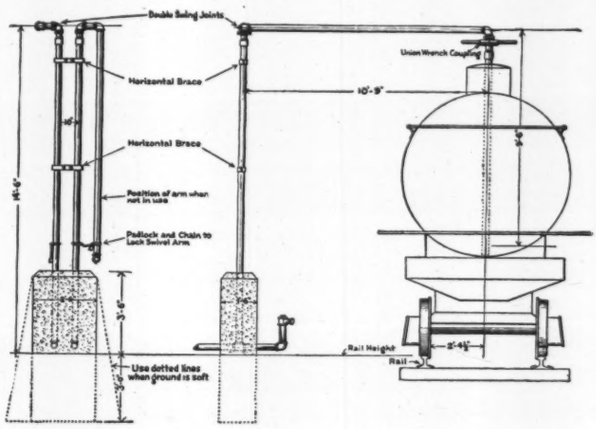


FIG. 7.—SHOWS THE DETAILS AND MEASUREMENTS OF AN APPROVED COMBINATION OVERHEAD AND UNDERNEATH UNLOADING APPARATUS FOR TANK CARS

which will permit the rapid escape of condensate but will not allow the escape of steam. When vertical pipe coils are placed in vertical cylindrical tanks, a good practice is to admit steam at the top of the coil and to discharge it at the bottom: by so doing the steam carries the condensate before it toward the trap, and eliminates the possibility of condensate remaining in the coils for any length of time.

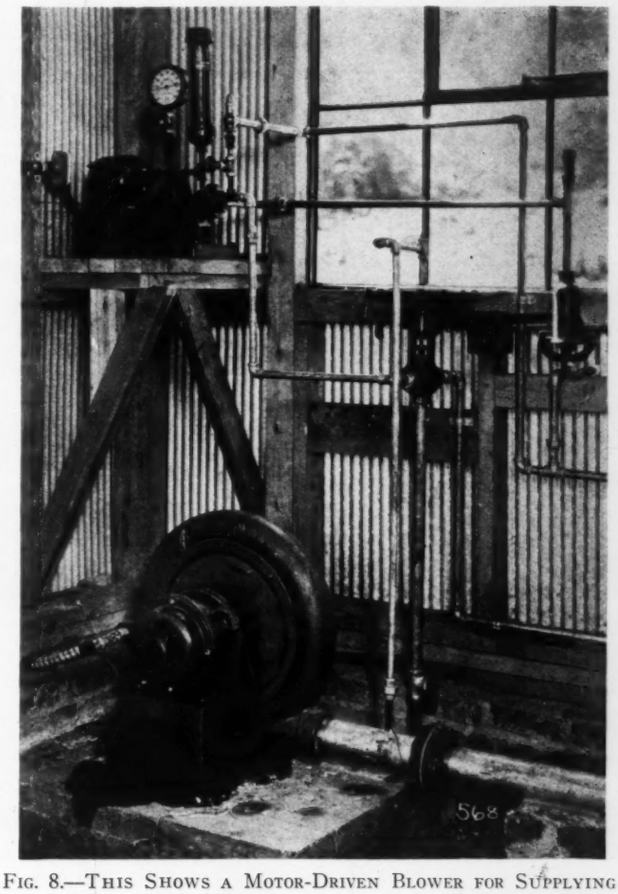


FIG. 8.—THIS SHOWS A MOTOR-DRIVEN BLOWER FOR SUPPLYING AIR TO LOW-PRESSURE BURNERS; IN UPPER LEFT CORNER IT SHOWS A MOTOR-DRIVEN, ROTARY PUMPING SYSTEM FOR SUPPLYING OIL UNDER PRESSURE TO THESE BURNERS, THE PUMPING SYSTEM EQUIPPED WITH PRESSURE GAGE, RELIEF VALVE AND COMPRESSION DOME FOR REDUCING VIBRATIONS AND PULSATIONS. AT RIGHT OF PICTURE IS OIL STRAINER ON SUCTION SIDE OF PUMP; TO LEFT OF STRAINER IS AUTOMATIC SHUT OFF

Whatever the type of coil installed in a fuel-oil tank, greatest economy will result if such coils are arranged in two separate units, one unit comprising just sufficient area so that the amount of heat introduced will balance that lost by radiation; the other, connected in addition

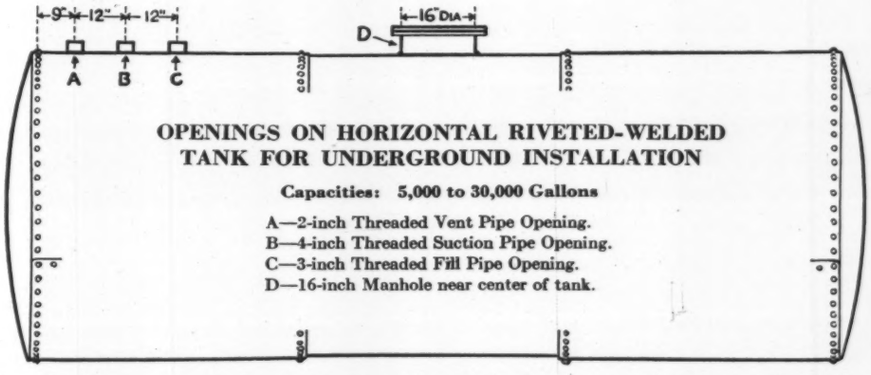


FIG. 9.—SHOWING THE OPENINGS ON A TYPICAL HORIZONTAL UNDERGROUND FUEL-OIL STORAGE TANK

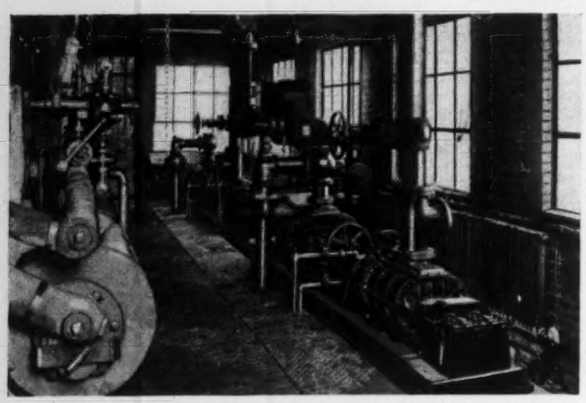


FIG. 10.—THESE BLOWERS WITH FUEL-OIL PUMPS ON IMPELLER SHAFTS FURNISH BLAST AND FUEL FOR ONE OF THE LARGEST ALUMINUM FOUNDRIES IN AMERICA

to the first, having such an area as to raise the temperature of the contents of the tank to the desired point in a given length of time. When heating the oil in the tank, the second unit should be "cut out" as soon as the required oil temperature has been reached, since it is more economical so to do than to throttle down at the valves the quantity of steam introduced. Such an arrangement also reduces trouble resulting from the collection of condensate, and better maintains the circulation of the oil in the tank.

Either live or exhaust steam may be used in the coils placed in fuel-oil tanks and, in some cases, where hot water is a waste product even this can be used to economic advantage.

In order to obtain satisfactory results in the storage and handling of heavy, viscous fuel oils it is absolutely necessary that the coils used in the tank be sufficiently designed and located. Keep in mind that the coils should be so placed that there will be a maximum natural circulation of the oil; also, that the installation must be such as to prevent any overheating of the oil whereby distillation would take place; and, further, that there must be ample room between the coil and the tank shell for the passage of the heated ascending oil and the descending cooled oil, which has been in contact with the outside tank walls.

When more steam than usual, considering the temperature, is required to heat the oil in the tank, it is probable the coils have been insulated by the formation of deposits that have settled out of the oil.

Fuel-Oil Piping

Pipe connecting a fuel-oil storage tank with the points at which the tank is filled or emptied should be perfectly tight and should run in the shortest possible direction. It is important that the number of turns and fittings in the line be at a minimum; for, the fewer the number of pipe joints, the less is the danger of leakage, and the fewer the turns, the less is the friction and subsequent loss in pressure. Pumps delivering to or from fuel-oil tanks should be valved on both suction and discharge sides, while pumps only delivering to tanks should be provided with a check valve in order to prevent the flow of oil from the tank back into the pump.

When long lines run above ground and are left filled with oil, the heat from the sun is often enough to cause expansion of the oil, then, if the valves at both ends of the line are closed, oil will be forced through the joints (either at the collars or valves) until the pressure is released.

In general, pump suction lines should be of larger size than the high-pressure discharge lines. This is especially advisable when the oil is thick and viscous, since the suction pressure tending to move the oil is very slight. Best results will be obtained if the pump is so

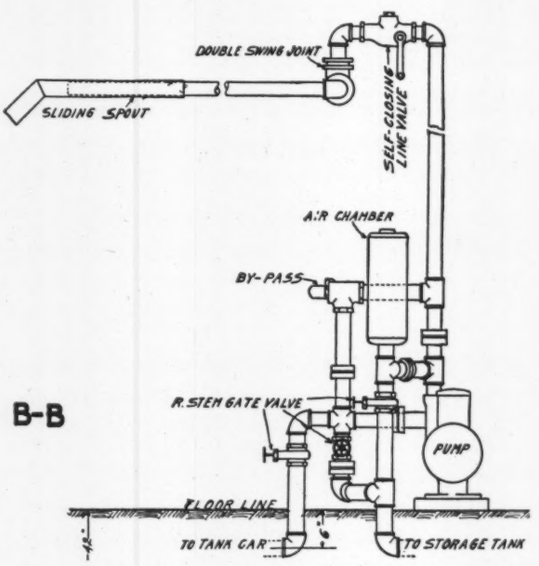
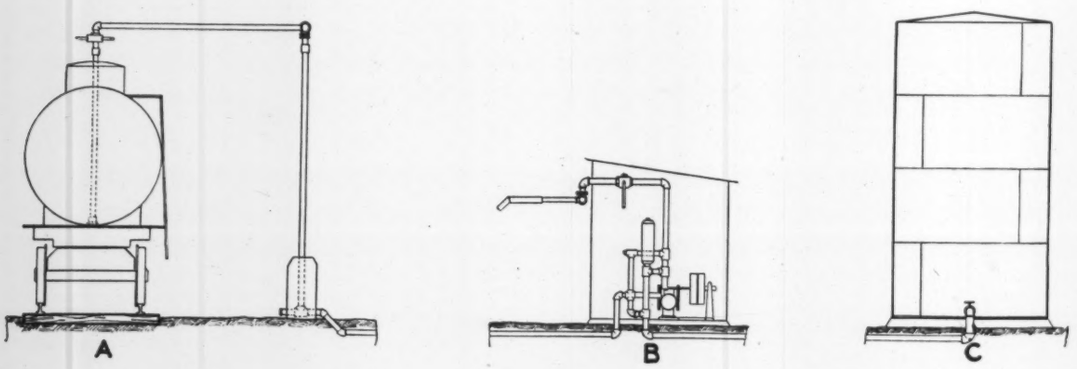


FIG. 11.—LAYOUT FOR PUMPING OUT A TANK CAR OF FUEL OIL TO STORAGE PUMP AND OTHER EQUIPMENT SHOWN IN B (DETAILS OF WHICH ARE SHOWN IN B-B) CAN ALSO BE USED IN DRAWING OIL FROM STORAGE



located that suction lines are very short and laid with a minimum of bends and fittings.

The formation of sediment in lines through which viscous fuel oils are pumped often presents numerous difficulties, unless provisions were made when laying out the system for the ready removal of such deposits. A proper strainer in the suction line will catch

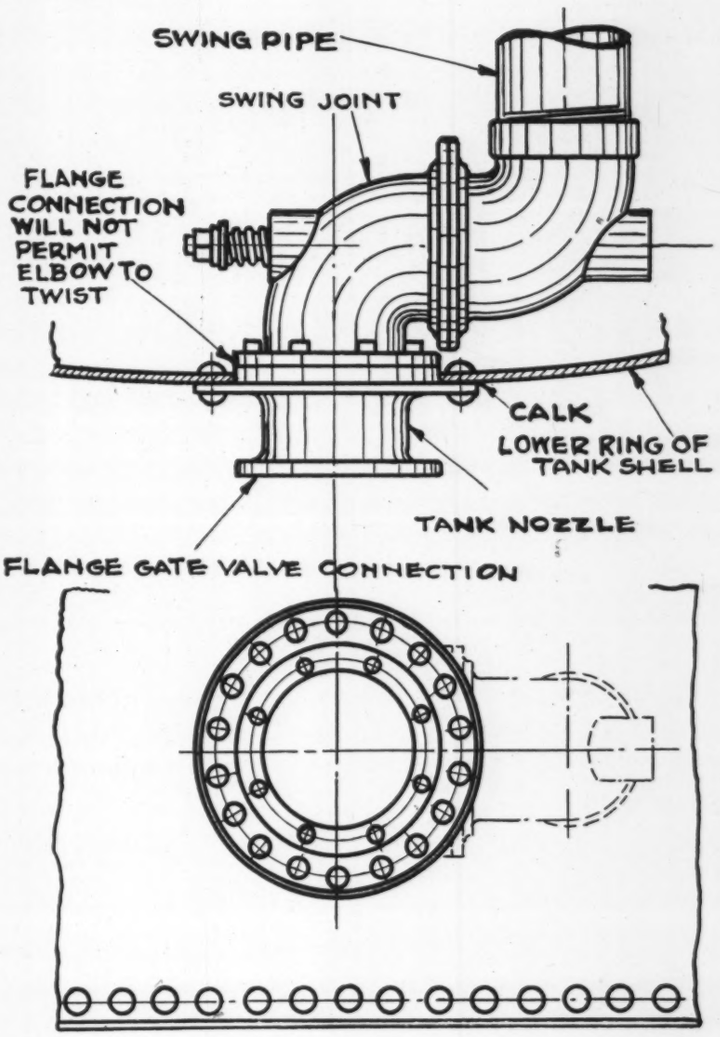


FIG. 12.—SHOWING DESIGN OF TYPICAL SWING JOINT FOR FUEL-OIL STORAGE TANKS. THE TWO HALVES OF THE SWING JOINT ARE HELD TOGETHER WITH A SUBSTANTIAL SHAFT ON WHICH IS FITTED A COMPRESSION SPRING. THE SHAFT IS LEAD PACKED AT EACH END TO PREVENT LEAKAGE

most of this sediment, if a swing line is not used, while, if a swing line is employed, fairly clean oil will be obtained, as almost all the solid impurities will have settled to the bottom of the tank and can be removed through suitable cleanout openings.

To make easier the removal of any sediment from lines, several suggestions are offered in this connection, when laying out a fuel-oil piping system: if no provision has been made for blowing out a long line with compressed air or steam, placing of occasional plugged openings in a line through which sediment or solidified oil can be removed by "rodding" or by running a wire through said opening is good practice. On a turn, use of a tee (one end of which is plugged), has the advantage over an elbow in that thereby it is

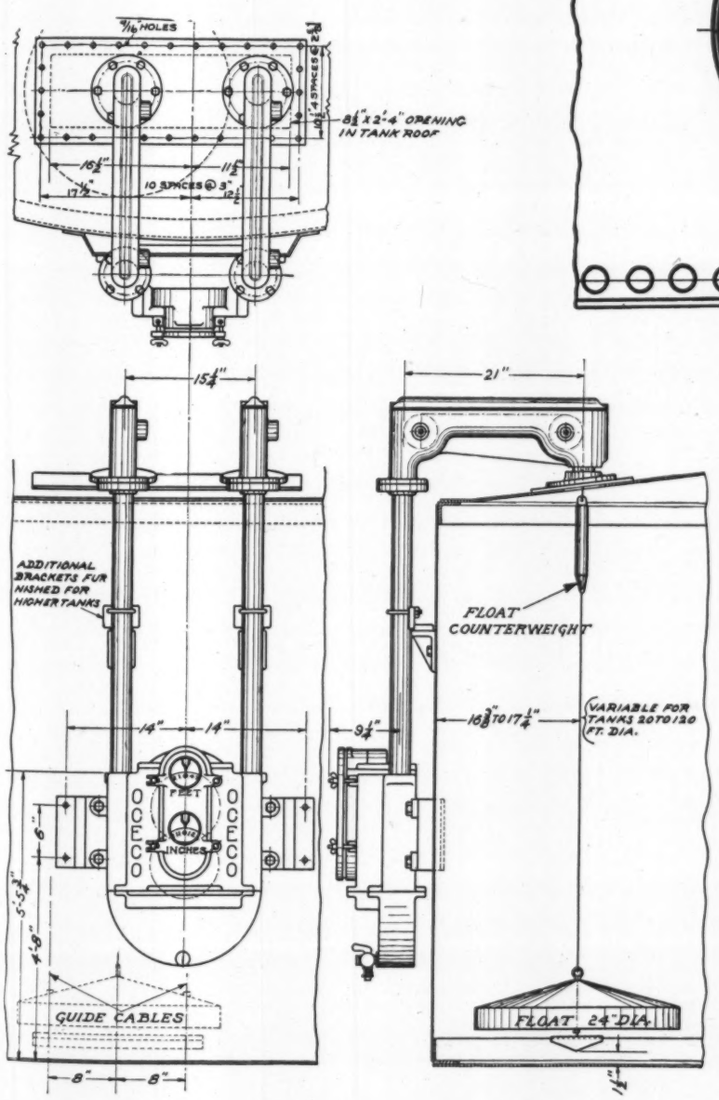


FIG. 13.—SHOWING DESIGN OF AUTOMATIC GAGE FOR FUEL-OIL STORAGE TANKS

possible to clean out dirt, etc., by rodding one branch of the line; while use of a cross, two ends of which are plugged, will give access to both branches.

On fuel-oil pipe lines of relatively short length, which can be opened by running a wire through same if there has been no provision for blowing out with compressed air or steam, pipe bends are more suitable than fittings due to the fact that they give less resistance to flow.

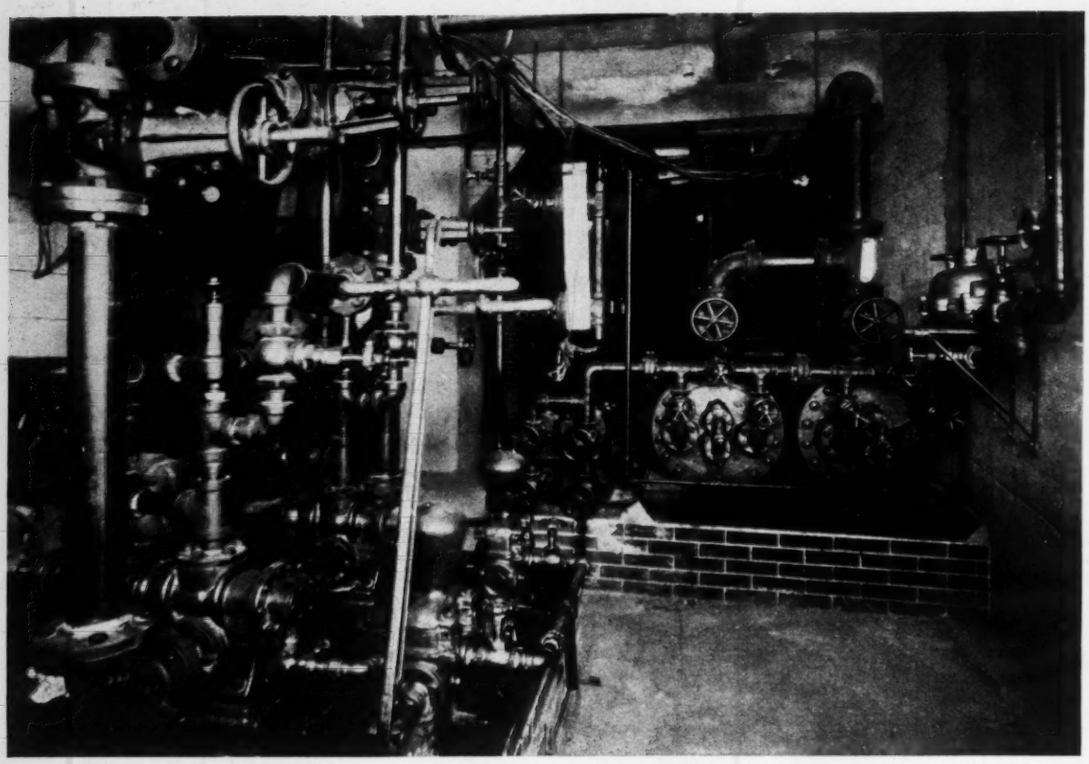


FIG. 14.—HIGH PRESSURE FUEL-OIL PUMPS AND PIPING IN BROOKLYN MUNICIPAL BUILDING, BROOKLYN, NEW YORK

If the oil to be handled contains any great percentage of constituents of low boiling point, lines should be laid out so as to prevent the collection in pockets of any gaseous vapors that may issue from the oil when it is heated, since the presence of such gas pockets will cut down the flow of the oil and pumping will be more difficult. By pitching horizontal pipes slightly upward in the direction of flow, danger of trouble from air or gas pockets usually will be eliminated; and possibilities of such trouble can be removed entirely by placing an air cock at the uppermost point in the system, opening this cock when necessary to permit the escape of air or gas. When leaks in the suction line introduce air into the system, such air must be disposed of through air cocks or into the air-cushion standpipe.

Heating Oil Lines

If fuel-oil lines are very long, or are run in the open where they are exposed to cold weather, it is generally advisable (especially with the heavier grades of oil) to run a steam line parallel with the oil line. The oil and steam pipes can either be placed together within a buried split tile pipe or a concrete or wooden trench, or else the steam line can be run inside of the oil pipe. The advantage of the latter method is that all the heat radiated is imparted to the oil and there is no direct radiation from the steam line to the outside air. Exhaust steam can be used for this purpose. Heat losses can be minimized by filling the trench with some commercial heat insulating material or can even be reduced by employing common, dry sand.

Steam piping that parallels fuel-oil lines (and which serves to heat the oil) should be given enough pitch so that it will drain readily, and should be provided with steam traps—large enough to take care of any surge of water that may take place in the lines—at intervals not exceeding 100 ft.

In the second part of this article, to appear in an early issue, fuel-oil pumps, oil heaters, strainers, etc., will be discussed, as well as theoretical considerations involved in laying out a fuel-oil pumping and piping system and similar information not given herein.

Piping for Heating with District Steam

By Sterling S. Sanford

THE downtown districts of many of our large cities are now heated with district steam. Piping under the streets delivers the steam to the customers' premises, just as other public utility services are supplied to buildings. The popularity of this service is indicated by the report of the National District Heating Association, which shows that in 1928 thirty-eight heating companies served over 14,000 customers with approximately 29 billion pounds of steam.

The characteristics of district heating service require that the arrangement of piping in a building should be somewhat different than for a boiler installation. Some of these characteristics are:

1. Steam enters the building from an external source and after being condensed in the heating system, the condensation is discharged from the building to the sewer or to the heating company's return mains.
2. The supply of steam is practically inexhaustible.
3. Steam pressure is maintained in the heating company's mains day and night.
4. In most cases, steam enters the building at a pressure greater than that required in the heating system.
5. The amount of heat used is determined by metering the steam as it enters the building or by metering the condensation before it is discharged from the building.
6. Heat in the condensation should be recovered before the water leaves the building.

Arrangement of Valves

The district heating company usually brings the service connection into the building and terminates it just inside with a service valve. At this point the building piping begins. If the service valve has screwed connections a flanged union should be installed near it so it can be easily removed when necessary. In most cases the steam pressure in the street mains is such that the customer must use a pressure reducing valve. Figures 1 and 2 show the usual arrangement of valves where the service enters the building. The house valve is the one used by the customer for turning his steam on and off. A gate valve is used for this purpose. It is shown equipped with a sprocket wheel and chain so that it can be conveniently operated from the floor. It is quite essential from the standpoint of economy that all valves be arranged for convenient operation so that steam can be turned off easily at night and at other times when not needed. The pressure reducing valve is generally of the lever type in buildings where vacuum pumps are used. In smaller buildings the spring-loaded type, which is not so easily tampered with by unauthorized persons is quite generally used. Care should be taken in selecting a reducing valve of proper size. Capacities vary with the type and make of valve. Tables of sizes and capacities given by the manufacturers should be consulted when selecting reducing valves. A by-pass around the reducing valve makes repairs possible without interruption of

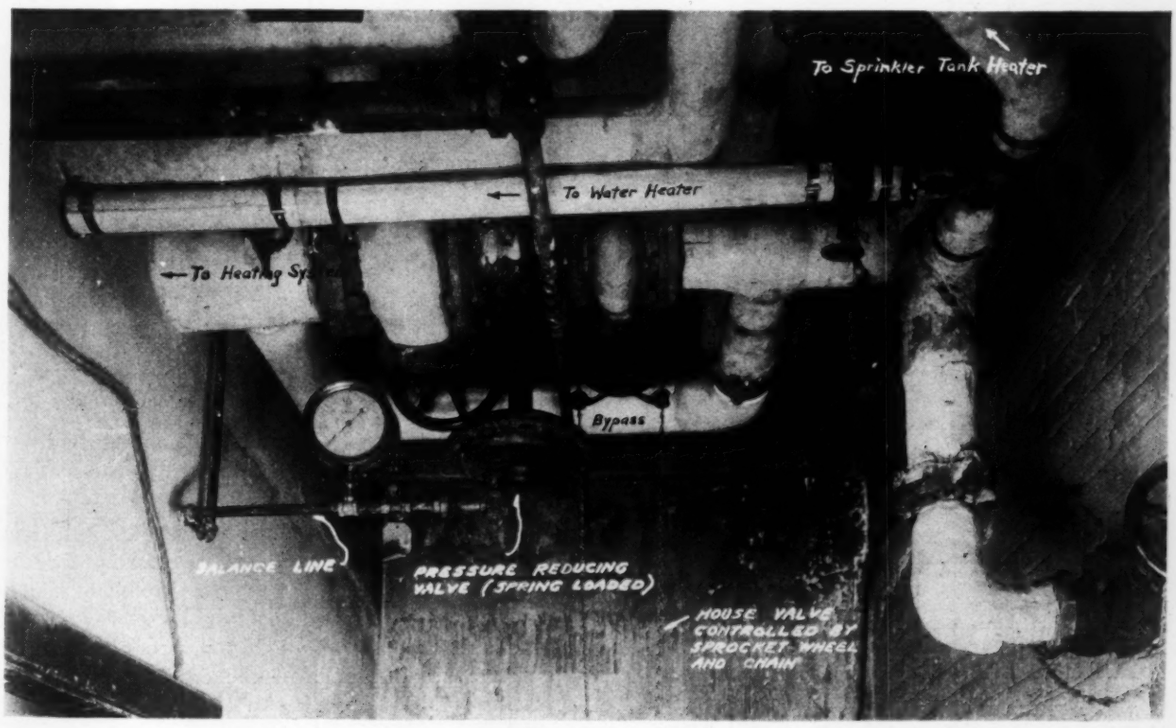


FIG. 1—PRESSURE REDUCING STATION WITH SERVICE ENTERING THROUGH BASEMENT WALL

service. Unions or flanges on either side of the reducing valve provide for its quick removal and replacement.

The Mains

Mains supplying steam to water heaters, sprinkler tank heaters, or to fan coils are taken off on the high pressure side of the reducing valve as the pressure required for such use is usually greater than that desired for the heating system. In buildings requiring heat for longer periods in some parts than others, the sections requiring long-hour heating should be served by separate mains so that steam can be turned off in the rest of the building without interfering with the parts requiring heat. Some buildings have places in them which must be heated for twenty-four hours a day, such as telephone switchboard rooms, lobbies in clubs and hotels, all-night drug stores and restaurants, and places for night watchmen. Installation of a twenty-four-hour main to serve these places makes it possible to turn off steam in the rest of the building. Savings of 35 per cent in steam consumption have been obtained by using such a main.

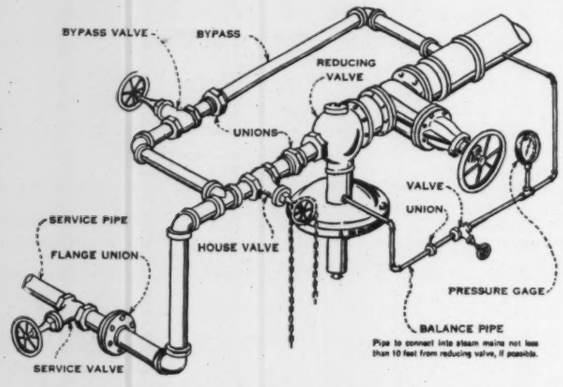


FIG. 2—ARRANGEMENT OF VALVES WHERE STEAM ENTERS BUILDING

Office buildings usually have first-floor shops which are open in the evenings. These shops should be served by a separate main. If the part of the building requiring long-hour heating is relatively small, the main supplying it should branch off ahead of the main reducing valve and be equipped with a separate small reducing valve. This will prevent damage to the seat of the large valve due to wire drawing at low loads. The heating systems in large buildings should be divided into zones to take advantage of variations in demand for heat in different parts of the building due to sun, wind or other causes.

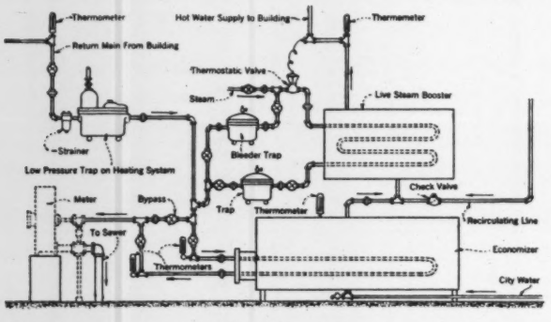


FIG. 4—ARRANGEMENT OF WATER HEATER, ECONOMIZER AND METER AT RETURN END OF A GRAVITY HEATING SYSTEM

In buildings such as store buildings where there is one service connection serving several customers, it is customary for the owner to install a steam header through the building from which the different stores can be served. Fig. 3 shows such an arrangement. The condensation in the header is piped to a separate meter and is charged to the building owner. Each customer must be able to control his steam supply conveniently. If his control valve is located in the basement, equipping it with an extension stem extending up through the floor adds greatly to the convenience. This is a necessity if the customer does not have access to the basement. As an alternative to the header arrangement, separate branches from the service connection to the individual stores can be used. In this case, however, the different customers must have access to their control valves near the service connection or else remote controlled valves must be installed which can be operated from the different stores.

Metering the Steam

The usual way of determining the amount of steam used is by a condensation meter to which all of the condensation is piped. The other way is to meter the steam with a flow meter as it enters the building. One large heating company uses flow meters extensively. Over 98 per cent of the meters used by other companies, however, are of the condensation type and some companies use condensation meters exclusively. Where a condensation meter is used, all condensation, including that through drip lines, must be returned to the meter.

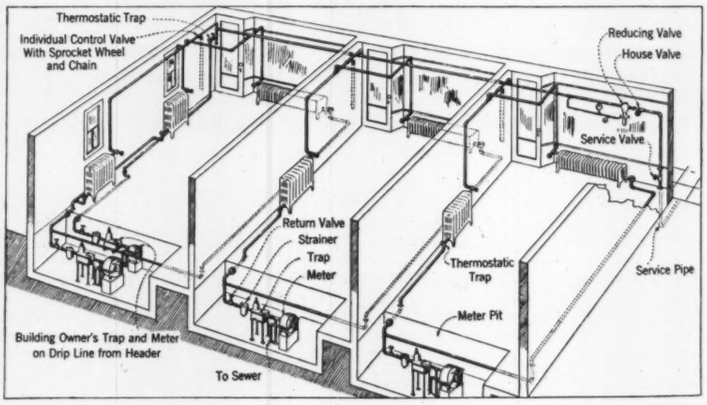


FIG. 3—STORE BUILDING SHOWING ONE SERVICE CONNECTION SERVING THREE CUSTOMERS BY MEANS OF A HEADER ACROSS THE FRONT OF THE BUILDING.

Underground Returns

Underground returns must be installed with care to prevent external corrosion and to make it possible to detect leakage if it should occur. Piping buried in the ground may rust quickly. For these reasons underground returns are often laid in crock with the hubs cemented and the ends of the crock left open so that leakage can be detected. An alternative may be to use cast-iron pipe with proper precautions to prevent external corrosion and leakage at joints.

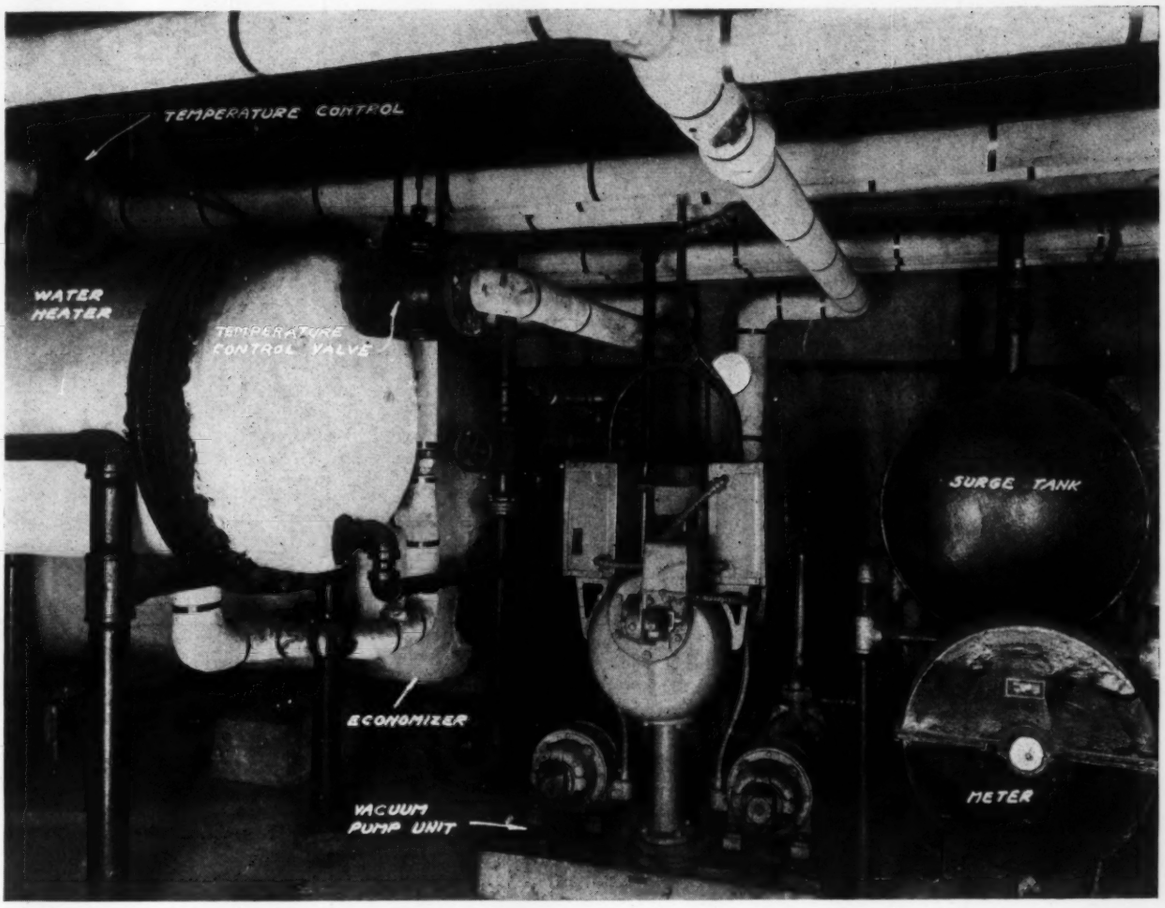


FIG. 5—VACUUM PUMPS, HEATER, ECONOMIZER AND CONDENSATION METER AT RETURN END OF A HEATING SYSTEM

Economizers

Fig. 4 shows the arrangement at the return end of a gravity system using a condensation meter. The water-heating economizer shown in the diagram is a means of recovering the heat in the condensation. The heat of the liquid is about 15 per cent of the total heat of the steam and the part of this recovered by an economizer is well worth while. The economizer consists of a water storage tank having coils in it through which the condensation passes. Cold water enters this tank on its way to the water heater and picks up heat, thus reducing the amount which must be supplied by the heater.

Fig. 5 shows the arrangement at the return end of a vacuum system. The following table gives the recommended heating surface in economizers for different sizes of office buildings. These sizes of economizers are the ones found to give the highest return on the investment.

RADIATION, SQ. FT.	ECONOMIZER HEATING SURFACE, SQ. FT.
1,000	10
2,500	30
5,000	60
7,500	90
10,000	120
15,000	180
20,000	240
30,000	350
40,000	450
50,000	550
75,000	810
90,000	970

In other types of buildings the size of the economizer must be determined from the amount of condensation available and from the hot-water requirements of the building. If the use of hot water is large compared with the amount of condensation, the effectiveness of the economizer is increased and its size can be reduced. On the other hand, if the use of hot water is comparatively small, it may not be possible to justify the investment in so large an economizer as recommended in the table.

Other steam-saving devices can usually be justified in a building heated with district steam and should be provided for in arranging the piping. Such devices include thermostats, remote-controlled valves, valves on risers, orifices in radiators, controls for turning steam on and off automatically, distant-reading thermometer systems, and vacuum pumps and reducing valves capable of operating through a wide range of pressures and vacuums. This applies particularly to district steam because the inexhaustible supply requires manual effort or else some means of control to prevent unnecessary use.

Tests of Protective Pipe Coverings

Because of the importance to the gas industry of the problem of underground pipe protection, the American Gas Association has supplied funds for the study of the problem and has appointed a research associate at the Bureau of Standards. The object of the research is to determine characteristics and properties of a coating material which is available and cheap and which will furnish the best protection to a pipe.

School Building Ventilation

By **Samuel R. Lewis**

I HAVE kept quiet on this subject for quite a while. It is about time I broke loose again. I am tired of being asked "What is new in school house ventilation?"

How do I know?

I know that I do not know much of anything about it, not being a biological expert.

If you want some opinions of an experienced dabster in school house ventilation, but not those of a biological expert, read the following conversation.

If you will write me your agreements or disagreements or just your opinions following your reading, look out, for maybe they'll print it in the paper!

Mr. Spafford: How important is the subject of school room ventilation?

Mr. Lewis: I think its importance has been exaggerated. Little Johnny spends only one quarter of five-sevenths of nine-twelfths of his early life in school rooms. It can have no terrifically vital effect on his general life it seems to me, compared to the effect on that life of the other three quarters of the school days and all of the Saturdays and Sundays and vacations and holidays. I think that if we keep Johnny out of drafts and keep his feet warm and prevent him from getting overheated while he is in the school rooms we will be keeping ahead of Johnny's general average.

Mr. Spafford: How important is it in terms of age of pupils?

Mr. Lewis: The tests we are making at Harvard University may have a bearing on that, though I have not picked up anything so far in my life to indicate that ventilation cares very much about the age of the persons ventilated, taking into consideration what I have just said.

Mr. Spafford: What is the relative importance in various latitudes?

Mr. Lewis: In the South where windows are wide open most of the time it requires force to sell ventilation. I have installed several mechanical ventilating systems in the Jackson, Miss., schools, alongside several others which have direct radiators and windows adapted to permit free air passage while keeping out the weather. The authorities like the latter kind the best. I believe that we have a struggle to outlive the custom of nature which destroys the thing not used. So many days are balmy as we go south that they will not operate elaborate ventilating systems except when heating positively is required. I sympathize with and endorse this attitude.

Mr. Spafford: How important is school room ventilation in terms of mental activity of pupils?

Mr. Lewis: Again, the time spent by the pupils in school rooms is such a small proportion of their total time that no clean-cut statistics have been published. I doubt exceedingly whether it ever will be possible to prove by mental activity standards that any particular quality of ventilation makes much difference.

Mr. Spafford: How important is school room ventilation in terms of health of pupils?

Mr. Lewis: The answer is not a clean-cut one. The New York ventilation commission, after careful observations of many schools in different cities and after exceedingly painstaking studies of racial tendencies, sex influence, etc., seems not to have found that any very definite conclusion can be drawn in answer to your question. The commission seems to believe in the provision, for ventilation, of much less volume of air supply per occupant per unit of time than has been furnished in days gone by. All ventilating engineers now recognize, of course, that ventilation is not satisfied by the mere introduction into a room of any given volume of air, but that distribution and control of that air must be practiced, and that in many cases, where distribution and control are nearly perfect, ventilation may be accomplished with a very material reduction from the old standards for delivery of air per minute.

Mr. Spafford: Is there any better reason for air conditioning in school rooms than in other buildings?

Mr. Lewis: In view of the fact that there are no generally accepted statistics which show harm from failure to humidify or to dehumidify or to cool school rooms, I should reply that there is no better reason for air conditioning in school rooms than in other buildings. Since we believe that comfort rather than actual illness and death is the stake, I am of the opinion that the workers of the world, who stay where they work somewhat longer hours each day than do the children, and who work six and seven days each week the year round, should, under an intelligent racial conception, be considered ahead of school children. The world has expended vastly more in educating these skilled workers than it has spent in educating the children.

It seems rather that the rational choice, if air conditioning must choose, would be to give it to the workers.

Mr. Spafford: What is the present status of the controversy between advocates of mechanical ventilation and advocates of window ventilation?

Mr. Lewis: There isn't any particular contro-

versy. Both the American Public Health Association and the American Society of Heating and Ventilating Engineers are seeking only to develop truth. I believe that both parties are becoming more tolerant. The happy idea of *controlled* ventilation seems a pleasant meeting ground. The observations made so far indicate to me that both sides were over-radical and I believe they approach a common ground for more freedom or tolerance in temperature, reduced volume of air per person, more independence from outside weather conditions and more appreciation of the fallibility of the human element.

Mr. Spafford: What is the cause of the widespread criticism of mechanical ventilation?

Mr. Lewis: I base my reply on personal observation and my answer is that ignorance is the cause. This means ignorance in the design and ignorance in the operation of these plants. We find more frequently than you might believe, an entire school faculty and patronage which has no complaint to make of mechanically - ventilated schools. Invariably we find that the plants with which these people are concerned have been well maintained and that they are operated effectively. Again the complaints against mechanical ventilation are not as widespread as the proponents of some new scheme would have us believe. Again they are nearly all from the teachers, and teachers, while admittedly intelligent, are notoriously not mechanically-minded. Just as many complaints are registered by the teachers against window ventilation and against other kinds of ventilation as are registered against mechanical ventilation.

We do find many mechanical ventilating plants which have been neglected and which are out of adjustment, but we find good satisfaction obtainable with most of them after a few minutes adjustment by a competent person.

Mr. Spafford: What is the cause of the complaint of stuffiness, drafts, dead air, etc., where mechanical systems of ventilation are used?

Mr. Lewis: Mostly I imagine the cause is due to the fans not being operated. However, all heating plants must be powerful enough to give service in the coldest weather, so that during all weather warmer than the coldest weather it is a serious task to hold the temperature in the rooms from getting too high. This difficulty of course exists with gravity heating and ventilating plants as well as with mechanical plants. Drafts are the result of attempts to get rid of this surplus mild-weather heat. There isn't any dead air. This term dead air seems to mean stagnant air. It is rather definite evidence when advanced as a complaint, that when the air was "dead," they had shut down the fans.

Mr. Spafford: Is it true that, based upon respiratory illnesses as shown by absenteeism, the window-ventilated schools show a better record than do the mechanically-ventilated schools?

Mr. Lewis: The New York ventilation commission believes that it has so shown in several instances. The evidence is not accepted generally, because of the comparatively short periods of occupation and because the problem of proof is exceedingly difficult, having to be discounted for racial stocks, sex, age, weather conditions, and other variable factors.

Mr. Spafford: Compare the older mechanical systems based upon the inadequate CO_2 theory and an air supply of thirty cubic feet of air per minute per person with modern air conditioning systems.

Mr. Lewis: The older school ventilating systems deduced that to blow thirty cubic feet of warmed or tempered outside air per minute per pupil into a room from one side-wall opening somewhere about eight feet above the floor, with an outlet flue, constituted ventilation. Such a lot of new, dirty, dry or damp or cold or hot outside air gave control-troubles, caused rapid fluctuations, and involved heavy operating responsibilities. The modern concept calls for the distribution within the room of the air supply so that the body of each occupant is placed in a slowly moving air current. Compensation is made for window chill. Dust is removed. Moisture may be added or subtracted. While the most modern air conditioning systems may still move as much as thirty cubic feet of air per minute per person, often not more than three to six of those cubic feet are new air from outside; yet the people are comfortable and happy.

Mr. Spafford: What effect has the volume of air per pupil per minute on the incidence of respiratory diseases?

Mr. Lewis: No one knows. There is some suspicion that if Johnny has damp feet the addition of ten cubic feet per minute to the air movement will cause his feet to get colder more rapidly, chilling him more and in some mysterious fashion causing him to be kept home tomorrow. I have not seen any convincing proof of this. I believe that it doesn't matter greatly. Johnny may have caught his cold on the way home when he ate that piece of candy he picked up from the floor, or he may have caught a sneeze from his neighbor in the movie or maybe Katie, who had a cold, kissed dear little Johnny.

Mr. Spafford: What have the American Society of Heating & Ventilating Engineers, Bureau of Mines, Harvard University and others done to provide information on this subject?

Mr. Lewis: They have all put their shoulders to the wheel by joining with the research laboratory of the American Society of Heating & Ventilating Engineers to determine in a logical and scientific way the basic information about ventilation.

Mr. Spafford: What are the essential factors in school room ventilation?

Mr. Lewis: No one knows definitely. The Guide says that they involve temperature, humidity, air motion and air duct.

Mr. Spafford: Who are the authorities on these factors?

Mr. Lewis: I cannot answer that question exactly. The old Chicago commission on ventilation studied the matter. The two eminent Drs. Hill, one in America, the other in England, studied it. Dr. Drinker of Harvard is deeply involved in practical research on the matter at present. Our Mr. Carrier and our Professors Willard and Harding have all been well-enlisted in the research. Mr. Houghten is an authority on all phases of the subject. Mr. Howard C. Murphy is outstanding in his studies of dust, as is Professor Rowley in his investigations of methods of measuring dust.

Mr. Spafford: How important is janitor instruction in mechanically-ventilated schools as compared with window-ventilated schools.

Mr. Lewis: The mechanical plants are simple: Any man of reasonable intelligence can be taught in a short length of time to open the inlets and outlets and to start the fans. The design and the original adjustment of mechanical ventilating systems for schools require some skill.

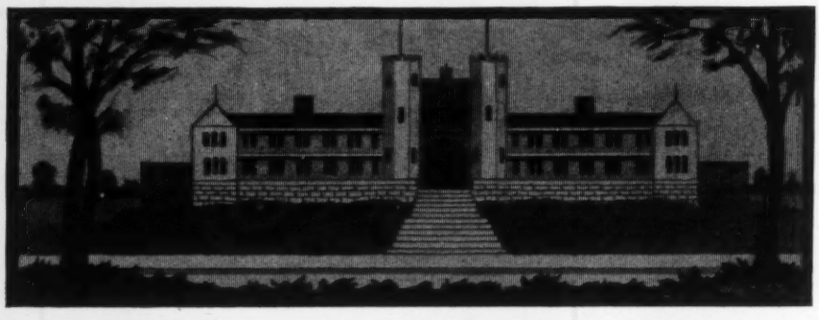
The quality of the men who are placed in charge of large schools in modern cities is undergoing rapid improvement. The engineer, of course, is in sole charge of the plant and the most difficult thing to make him understand thoroughly is that no serious consequence follows the opening of as many windows as the teacher wishes, at any time.

The window-ventilated schools have boilers, pumps, radiators, automatic temperature control, etc., just as do the mechanically-ventilated schools. They have exhaust fans for toilets, for kitchens, for laboratories, etc. They have everything which the ordinary mechanically-ventilated school has, except the supply fans and the motors which drive these fans. The engineer, however, cannot operate the windows. The teachers have to learn to operate them, as dictated by varying wind and storm and sunshine, and the teachers, as observed by myself, are prone to forget the windows and to subject the pupils to wild fluctuations of temperature, without any possible attention to moisture conditions, with gales of cold air from the windows striking some of the pupils, interspersed with much dust from the adjacent streets and sidewalks.

I conclude that there is no advantage in either side of the argument as to janitors.

Mr. Spafford: What are common errors in design of schools from the point of view of the engineer concerned with the heating and ventilating of such buildings?

Mr. Lewis: The architects usually are very liberal to the engineers. If the engineer does not get insulation on an exposed stage back wall, for instance, it usually is his own fault for failing to ask. If the building cannot be kept warm in a gale of cold wind because of leaky windows, the engineer should have forefended against the design or acceptance of those windows.



I believe that part of the duty of the engineer is to know enough of the virtues of building insulation to sell such insulation to the architect and the owner.

One of the persistent and very grave mistakes frequently made by architects is to place the ground story of the building directly on the wet damp earth, without any air space under the floor. Waterproofing the floor when it is so placed does not serve to stop the rapid heat loss with its tendency to cold toes on the part of Johnny and his elderly teacher.

Architects think that chimneys are unbeautiful and are prone often to crowd down the height of our smoke flues. The heating engineer sometimes is too good-natured concerning this matter. Exhaust outlets through roofs do not work well unless they look up. If they look out through the face of a wall, some day a wind will look in at that opening and that day the world will not know of the existence of the gentle little fan which is supposed to cause the air to go out. I know of one great vent shaft serving a laundry eighteen stories down, which goes democratic whenever a pleasant wind from the north looks into the north-facing discharge louvers. At such times the laundry help begins to faint and the ironing boards become stretchers.

Sometimes the architects try to make the engineers feed air into an auditorium from outlets which look toward the stage. This method of causing neck-back drafts always gets all concerned, the engineer, the architect and the neck owner, into trouble.

Mr. Spafford: Would the inter-relation of temperature, humidity, air movement, and air cleanliness be controlled automatically in the ideal school house ventilating system?

Mr. Lewis: There is no definite answer. Uniform temperature and uniform other conditions are never conducive to the best human environment as far as I know. Change is stimulating. I do believe, however, that we cannot leave temperature-control to chance.

I have been interested in recent thoughts concerning clothing. The present generation of young women, in my judgment, is superior physically to the young men. I suspect that this superiority is due to the very much more active bodily temperature-control functions of the scantily-clad woman as compared to the wool-clothing-swaddled man. We undoubtedly do wear too many clothes when indoors during the heating season and we should speed up our appreciation of the superior physical tolerance to temperature variations, and to adverse meteorogolical conditions, of the young half-dressed women of today so that the men also may partake of the new freedom. We should also provide controlled variations in indoor temperature.

There is no definite scientific answer to the humidity control question. It seems reasonable to believe that a point approaching saturation of the air every night during the heating season would be delightful, since we go through that damp period every summer night and suffer no ill if we keep from being bitten by a certain kind of mosquito. Dead uniformity in anything having to do with human environment is contrary to human desire and human endurance, so I suggest controlled fluctuation in all these factors of temperature, humidity and air-motion.

I think, however, I'd try to keep the cleaning factor clean all the time!

We know very well that, dealing as we are with very potent quick-acting agencies, we get in trouble at the extremes if we let them run wild. Why not intelligent temperance rather than license or prohibition in such matters?

The Design of Flanged Pipe-Joints—III

Waters-Taylor[1] Method for Computing Maximum Fiber Stress in a Loose Flange, Using St. Venant Criterion for Failure

By Sabin Crocker[2] and Arthur McCutchan[2]

(Continued from September Issue)

THE method described by Waters and Taylor in their paper on "The Strength of Pipe Flanges" applies to loose flanges with or without hubs. A rational formula for the design of plain ring flanges without hubs was developed, using as a basis the general theory of bending of circular plates. In the case of hubbed flanges, however, it was necessary to combine the theory of bending of circular plates with that of bending of tubes, to obtain an empirical formula which apparently is borne out by test results.

While the Waters-Taylor formulas are not directly applicable to fitting flanges, that for hubbed flanges might be applied by assuming an infinitely high hub, or, in other words, dropping out the term which has hub height in its denominator. (See Equation 2.) This assumption is probably warranted on the grounds that successive increments in hub height produce diminishing returns in flange strength, since a height is soon reached beyond which further increments give no significant increase in strength.

The maximum strain (St. Venant) theory was used by Waters and Taylor as the strength criterion for the design of loose flanges. This theory holds that, when a material is subjected to the action of two or more stresses on mutually perpendicular planes, failure will occur when a certain given deformation or strain is produced. This deformation being a function of the stresses acting, Poisson's ratio, and the modulus of elasticity of the material.

In deriving the formula for ring flanges, the flange was considered as a flat plate with a central hole, loaded near the outer and inner edges. The actual loading of the bolt forces acting on the bolt circle was converted to this assumed loading by the principle of equivalent couples. The analysis of a plain ring with this loading may be found in recent works on applied elasticity.[3]

The extension of the formula to include hubbed flanges involved the theory of the bending of a tube caused by a bending moment uniformly distributed around the end of the tube. (See Fig. 15.)

This moment is induced by the cupping action of the flange under load. The resistance offered to this moment by the rigidity of the tube wall or hub accounts for the well-known stiffening effect of a hub.

Since the only function of the hub portion is to supply this resisting moment at the inner edge of the ring, it is evident that there will be a point at which additional height of hub will not materially affect the strength or rigidity of the flange. The diminishing effect of increase in hub height is shown graphically in Fig. 16 for a 12-inch flange of the 600-lb. American Steel Flange Standard.

In applying the formulas to the design of flanges, the bolt load W in the formulas is most conveniently expressed in terms of tension in the bolting material. A definite assignment of stress is unnecessary, as the flanges should be proportioned in accordance with the expected bolt load as previously explained in the introductory paragraphs in Part II of this article, appearing in the September issue.

The next point involved concerns the location of the deforming couple. In applying the Waters-Taylor formulas the bolt forces have been considered as a uniformly distributed load acting on the bolt circle. The gasket reaction which completes the couple tending to cup the flange is considered as concentrated on a circle midway between the inner and outer edges of the contact surface. This assumption of loading gives the greatest lever arm and consequent moment to which the flange may be subjected, and appears to be the logical basis for design with the Waters-Taylor formulas. A careful analysis of loose flanges of the American Steel Flange Standards, using the Waters-Taylor formula for hubbed flanges and with the loading assumptions stated above, indicates an average stress ratio of approximately 0.85.[4] The shifting of the points of application of the bolt loads toward the inner edges of the bolt holes and the like shifting of the gasket reaction toward the outer edge of the contact surface, noted in the foregoing discussion of methods of flange calculation, gives an added factor of safety which may reduce the stress in the flanges from 25 to 50 per cent. Credit should not be taken for this reduction in the acting moment as the flange deflects, unless a stress ratio correspondingly lower than 0.85 is set as a basis for safe design. In any event, a stress ratio should be maintained which makes due allowance for the relative physical properties of bolt and flange material, all factors considered.

The lever arm (a) of the formula becomes in the case of a Van Stone flange the distance from the bolt circle to a circle midway between the inner and outer

Note 1: See A.S.M.E. paper on "The Strength of Pipe Flanges," by Everett O. Waters and J. Hall Taylor, published in the mid-May, 1927, issue of Mechanical Engineering.

Note 2: Engineer, Engineering Division, The Detroit Edison Co.

Note 3: See Applied Elasticity, by S. Timoshenko and J. M. Lessels, page 286.

Note 4: As explained in a preceding paragraph, the term stress ratio is $\frac{\text{maximum stress in flange material}}{\text{tensile stress in bolt material.}}$

edges of the contact surface between the flange and the lapped over end of the pipe. The inner edge of the Van Stone flange is rounded to allow for the fillet at the base of the pipe lap. This rounding is usually done to give a radius of ⅜ inch so that the actual surface in contact is reduced by this amount.

The formula for determining stress in plain ring flanges as given on charts prepared by Waters and Taylor is

Equation 1—Plain Ring Flanges

$$s = \frac{5.72\ Wa}{T^2(A-B)}\left[\frac{A^2 \log A/B}{A^2 - B^2} + 0.117\right]$$

In which

s = maximum stress, lb. per sq. in.
W = total bolt load = total cross sectional area of bolts at root of thread times tension in the bolts.
a = distance from circle on which bolt forces act to circle on which reacting forces are concentrated, in.
T = flange thickness, in.
A = outside diameter of flange, in.
B = inside diameter of flange, in.

The formula for plain ring flanges is simple in form and application but is not of as great importance for general flange design as that for hubbed flanges since the advantageous distribution of metal in hubbed flanges has rendered them a more popular type. The formula for hubbed flanges expressed in its final form is as follows:—

Equation 2—Hubbed Flanges

$$s = \frac{5.72\ Wa}{B(K-1)T^2}\left[\frac{K^2 \log_{10} K}{K^2 - 1} + 0.117 - \frac{K^2+1}{K^2-1}(X)\right]$$

In which X is given by the relation

$$X = \frac{3.33\ K^2 \log_{10} K + 0.389\ (K^2-1)}{4.33\ K^2 + 2.33 + \left[(K^2-1)\ T^3\left(\frac{0.084\ B^2}{h^5} + \frac{1.56}{\sqrt{B\ g^5}}\right)\right]}$$

In which nomenclature is as given above for ring flanges with the exception of dimension B and the additional terms defined as follows:—

K=A/B ratio.
A=outside diameter of flange, in.
B=diameter to center of hub, in.
h=height of hub plus half flange thickness, in.
g=thickness of hub, in.

These formulas appear rather cumbersome but the same terms are repeated so frequently in only slightly different relation that the solution is actually quite simple, especially if made with a slide rule.

Typical Calculation

The following typical calculation for the stress in a 12 inch, 600 lb. American Standard Steel Companion Flange should clarify any obscure points in the explanation and demonstrate how to calculate stress in hubbed flanges by means of the Waters-Taylor formula.

The first step is to make a table of the necessary data from which the factors involved in Equation (2) may be obtained. All of the dimensions save the diameter and height of hub are given in the "Tentative American Standard for Steel Pipe Flanges and Flanged Fittings." (B 16e—1927.) The latter were taken from the Proposed American Standard for Steel Companion Flanges. (A.S.A. B 16b2—1929.)

12 inch 600 Lb. Steel Companion Flange (See Fig. 16)

K=A/B=1.542	a=2.50	T=2⅝	Diam. Hub 15¾
A=22	g=1.50	20—1¼-in. bolts	Length through Hub 4⅝
B=14¼	h=3.3125	W=18.60 t	Dia. Raised Face 15

Diam. Bolt Circle 19¼ in., O.D. Pipe 12¾ in., Diam. inner edge of contact circle 13½ in.

These values are then substituted directly in Equation (2) giving the rather complicated appearing expression:

$$s = \frac{5.72\times18.60t\times2.50}{14.25\times0.542\times6.89}\left[\frac{2.38\times0.1885}{1.38} + 0.117 - \frac{3.38}{1.38}X\right]$$

In which the value of X is given by the relation

$$X = \frac{(3.33\times2.38\times0.1885) + (0.389\times1.38)}{(4.33\times2.38) + 2.33 + \left[1.38\times18.1\left(\frac{0.084\times203}{400} + \frac{1.56}{\sqrt{14.25\times7.60}}\right)\right]}$$

$$s = 5.00t\ (0.325 + 0.117 - 0.285) = 5.00t\ (0.157)$$

$$s = 0.785t$$

The stress ratio of 0.785t determined for the above flange is in line with the average relation between flange stress and bolt tension of 0.85t which is found to exist in American Standard Steel Companion Flanges, when analyzed by means of the Waters-Taylor formula, and is on the side of safe design.

The relation between flange stress and bolt tension determined for this same flange, using the method described in connection with Fig. 11 in Part II of this article, which appeared in the September issue of this publication, is 1.13t. The Waters-Taylor formula gives a stress ratio approximately 30 per cent less for this particular flange. While the stress ratios are of different magnitudes, it is found that a series of flanges proportioned in accordance with either method will differ in only one essential respect; viz., height of hub.

In the case of the 12 inch 600 lb. flange under investigation a hub height of 3½ inches would be required to give a stress ratio of 0.85t using the method based on the Rankine criterion of failure.

The application of the Waters-Taylor formula, while tedious, is not at all difficult as it involves only the simplest algebra. The results seem to be in excellent agreement both with test data on hubbed flanges and with actual cases of flange failures and successes which have accumulated throughout years of practical experience with pipe flanges.

The Sub-group on Loose Flanges of the A.S.A. Sectional Committee on Pipe Flanges and Fittings has made use of the Waters-Taylor formula for

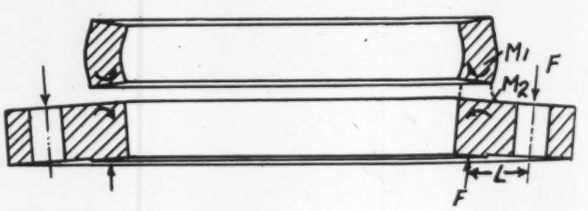


FIG. 15. DEFORMATION OF HUBBED FLANGE. M_1 IS MOMENT INDUCED IN HUB BY THE DEFORMING MOMENT FL. M_2 IS THE EQUAL AND OPPOSITE MOMENT SUPPLIED TO THE FLAT RING PORTION OF THE FLANGE BY THE HUB, WHICH TENDS TO PREVENT CUPPING OF THE FLANGE UNDER LOAD

hubbed flanges in working out the design of companion flanges for the Steel Flanged Standards. The outside diameter of hub was first determined for each size of the standards so that box wrenches might be conveniently used with pipe sizes 4 inch and larger and bolts 1 inch diameter and larger. (See Fig. 9 in Part I of this article, which appeared in the August

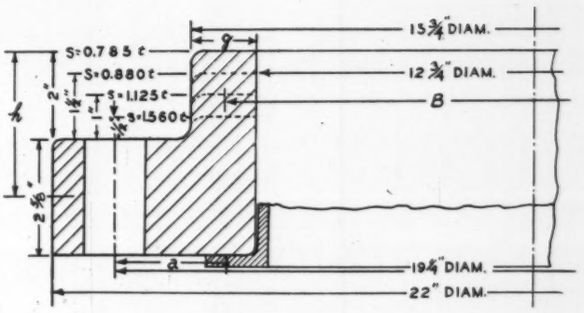
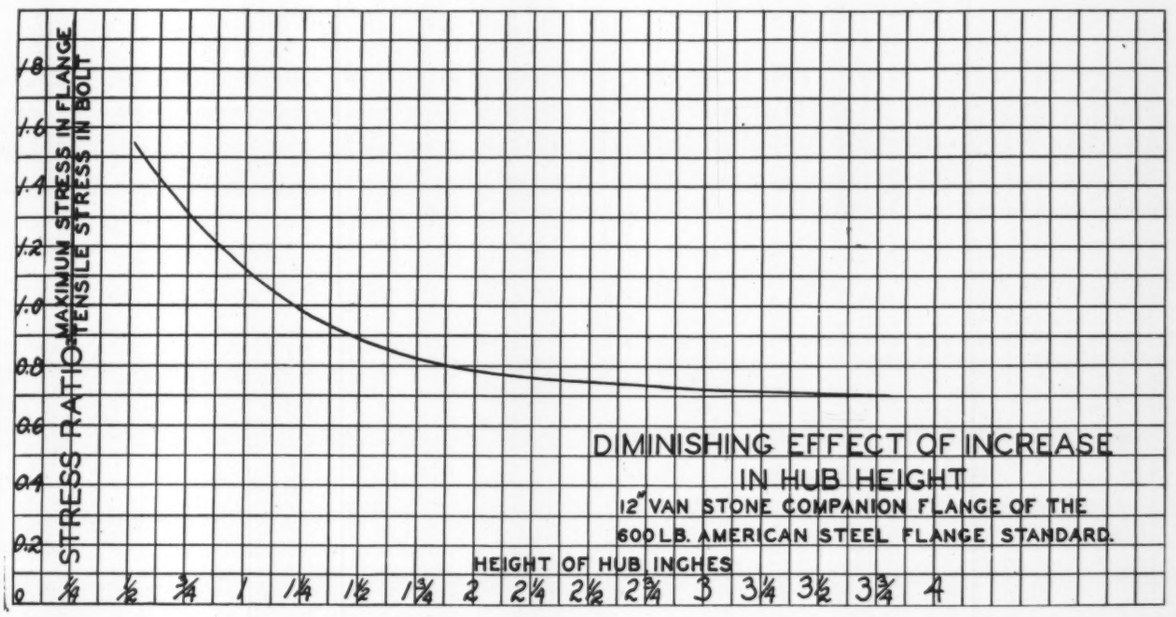


FIG. 16. DIMINISHING EFFECT OF INCREASE IN HUB HEIGHT


STRESS RATIO=MAXIMUM STRESS IN FLANGE / TENSILE STRESS IN BOLT
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
DIMINISHING EFFECT OF INCREASE IN HUB HEIGHT
12" VAN STONE COMPANION FLANGE OF THE 600 LB. AMERICAN STEEL FLANGE STANDARD.
HEIGHT OF HUB INCHES
¼ ½ ¾ 1 1¼ 1½ 1¾ 2 2¼ 2½ 2¾ 3 3¼ 3½ 3¾ 4

issue of this publication.) The height of hub was selected in accordance with the diminishing benefits of successive increments in hub height as explained in a previous paragraph in connection with Fig. 16.

With these assumptions of hub dimensions and loading and with the flange dimensions and bolting given in the Steel Flange Standards, it was found that the majority of the flanges have a stress ratio of approximately 0.85 the bolt tension.

Editor's Note: This concludes "The Design of Flanged Pipe Joints," the first part of which appeared in August and the second part of which appeared in September.

Guarding the Heating System

A guard against freezing in the heating system of the Smith Engineering Company of Milwaukee, Wisconsin, has been put in in the form of a gasoline driven centrifugal pump. The offices and plant of the Smith concern are heated by a hot water system, the proper circulation of water being produced by means of two electric motor driven centrifugal pumps. The electric driven system works efficiently while the current is on, but should the current be turned off, or a fuse be blown, or the pumps from any other reason cease to operate, the entire heating system immediately becomes in imminent danger of freezing and crackage.

The first experience which this company had with freezing was during the season of 1928, when a heavy storm came up over the week-end, and the current was off for a period of some nine or eight hours. The result was that an entire section of the cast iron boilers which were on the job at that time was broken and required replacing. During the same winter the current was off a second time, but the gasoline centrifugal pump had been purchased and no damage was done. The gasoline driven centrifugal is directly connected to the main line, and will circulate the water throughout the entire heating system in case of an emergency.

Date Set for I. A. A. Convention

Announcement has been made that the annual convention of the International Acetylene Association will be held at the Congress Hotel, Chicago, November 13, 14 and 15. This annual meeting is an outstanding event in the oxy-acetylene field each year and the programs from year to year have furnished valuable and significant contributions to the literature of welding.

The program committee has made definite arrangements for a number of papers of unusual interest and it is expected that some detailed information on this subject will be available very soon. However, Mr. L. F. Loutrel, president of the association, has advised all members that suggestions for making the program more complete and more constructive will be gladly received. Those interested in the promotion of oxy-acetylene welding and cutting can be sure of finding both pleasure and profit in attending the coming meeting.

INTERVIEWS OF INTEREST

Pittsburgh Makes Observations on Atmospheric Pollution

Much impetus has been given recently to the study of air pollution by the participation of American Society of Heating and Ventilation members in dust sampling work sponsored by the American Public Health Association. It is thought that the tabulated observations from different parts of the country will give valuable comparisons of conditions of air pollution in typical industrial cities and will point the way for control of dust pollution at its source.

It is of particular interest that in Pittsburgh, where most of our literature on air pollution by combustion products has arisen, the dust collecting and counting has been expanded into a major piece of basic research. The Pittsburgh work is sponsored by the Mellon Institute under W. A. Haomr, assistant director, and H. B. Meller, senior industrial fellow of the Institute. Mr. Meller, a combustion engineer, works through the department of industrial wastes, Pittsburgh health department.

Taking Dust Samples

"A single dust count in a sheltered situation made once in twenty-four hours may furnish interesting basis for general comparison from city to city," states Mr. Meller; "but it does not begin to furnish the fundamental information that is essential for effective control. The Pittsburgh plan carries twenty-five dust collecting stations, one for every two or three square miles throughout the city.

"It is known that the quality of air, and the intensity of sunlight differ very greatly within an area of a few dozen city blocks. We want to study the etiology of dusts, their kinds and their sources, and their specific influence upon disease.

"Dust samples taken every twenty-four hours will be correlated with temperature, barometric readings, and surrounding industrial activities. It will be seen what relationships exist between atmospheric impurity and fogs, for instance, and what peaks and falls hinge upon cessation of factory processes on Sundays and holidays. Other samples will be collected once in thirty days, and these will give the quantities necessary for full chemical study of the nature of the precipitates.

Microscopic Dust Counts

"This relates to heavy dusts that tend to fall close to the site of their origin. Another phase of our work will deal with microscopic dusts that remain suspended in the air. We need to know how generally such microscopic dust acts to filter out of city air the particular wave lengths of light man needs for health. Certain dusts cause or intensify fogs. It is suggested also that certain organic dusts serve as nuclei collecting contamination and conveying disease.

"This part of the work will be subjected to a double check. The microscopic counts first made by means of Owens' jet dust counter will be verified by use of the calibrations calculated at the laboratories of the Society by Margaret Ingels. At this point, important advances are looked for in the development of a photoelectric cell for microscopic work, capable of precise, continuous record, and at the same time reliable in the hands of persons not rated as expert observers.

"A third phase of the Pittsburgh study is concerned with solar radiation. If the air of our cities lacks ozone because the microscope dusts cause selective light interference, what percentage of dust filters out the health-producing ultraviolet light, and what is the means of control?

"We utilize here a perfectly new precision instrument, a spectrograph recently developed by Dr. H. C. Renschler, at the Westinghouse laboratories. We are also working out a new photoelectric cell for the measurement of light.

Discover Screening Effect

"Our work on light will include the entire spectrum. We believe that the present preoccupation with ultraviolet light will soon spend itself and that competent work is called for on infra-red, and other parts of the spectrum. Our data will aim to discover more than the mere fact of exclusion of ultraviolet light. We want to show just what the screening effect is of prevailing types of dust. We have effective means at hand for precipitating metallic dusts. Is it ethically necessary and economically possible to remove organic dusts? Whole populations benefit from the removal of the ruinous hydrogen sulphite from city air. When we know what menace a given dust represents and where it arises, we will be in position to prescribe equally practical and helpful methods of control."

Institute Has Made Other Dust Studies

This is not the first time that the Mellon Institute has made history in the study of atmospheric pollution. It was during 1911 to 1914 that the Institute laid down the first broadly scientific study of air pollution in Pittsburgh. After a suitable interval following their measures for control, another study in 1923-1924 brought out the important fact that only fume pollution had decreased. Total solids had steadily increased in percentage. Ten bulletins and a number of papers widely published bear records of this earlier work.

The present correlations of qualitative and quantitative analysis of dust with barometric, health, and other conditions that relate to dust, whether visible or invisible, will establish standards for future work.

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Meets: Third Monday in Month
President, W. F. NOLL
1188-48th St.
Secretary, V. A. BERGHOEFER
1640 Holton St.

Pressure Difference Across Windows in Relation to Wind Velocity

By J. E. Emswiler,[1] Ann Arbor and W. C. Randall,[2] Detroit, Mich.
MEMBERS

THE amount of air passing through a window as leakage is dependent upon the pressure difference across the window, and the crack opening. At the present time, there is a considerable mass of data from laboratory research showing what the leakage is for various kinds of windows for any specified pressure difference; but there is little definitely known about what the pressure difference actually is or is likely to be in any given building. In the absence of knowledge regarding this quantity, it is usual to assume a certain wind velocity appropriate to a given locality, and take the leakage as that corresponding thereto, which is equivalent to assuming a certain pressure difference at the window.[3]

Tables showing the relationship between chosen wind velocities and window leakage per foot of crack that may be expected to occur with pressure differences corresponding to those wind velocities, are given on pages 51 to 54 of THE GUIDE 1929 for various types of windows.

The principal object in predetermining window leakage is to enable due allowance to be made in the amount of heating surface for the expected maximum infiltration. This maximum does not mean the leakage that will occur with the highest momentary gust of wind, nor the highest rate for any one hour, but should rather be considered as that which may occur during several hours of the heating season.

It is stated in THE GUIDE that "the heat allowance for infiltration through cracks must be based on the average wind velocity for a given locality." It seems probable that the *average wind velocity,* as the term is used there, does not mean the actual average for all the hours of the heating season. In the case of transmission losses, THE GUIDE suggests 15 miles per hour as an *average wind velocity,* and it is probably implied that a similar value is intended to be used in the calculation of infiltration losses, in the absence of specific data. It is also stated in THE GUIDE that "a further allowance must be made for the direction of the prevailing wind in any locality, which shall be done by adding 15 per cent to the infiltration losses on the sides of the building exposed to the prevailing winds." The tables of THE GUIDE are given in two parts, the values of Part II being 80 per cent of those for Part I (in which leakage as determined from experiments is given), to make allowance for an opposing pressure built up on the inside as a result of air being forced in by the excess wind pressure on the outside.

In a former paper by the authors,[4] the need for more definite information on the subject of pressure difference across windows was emphasized, and it was suggested that a program of work along this line should be started. It was with this thought in mind that the study described in the present paper has been undertaken. The definite objectives are to attempt to show how the chosen value of wind velocity for THE GUIDE Tables should be rationally determined for a particular case; to ascertain if the factor of 80 per cent of Part II is a reasonable value; and, if possible, to see what is the effect of temperature difference. It is to be understood that this study is by no means complete, but is offered at this time rather more in the hope of receiving criticisms and suggestions regarding the mode of procedure in the inter-

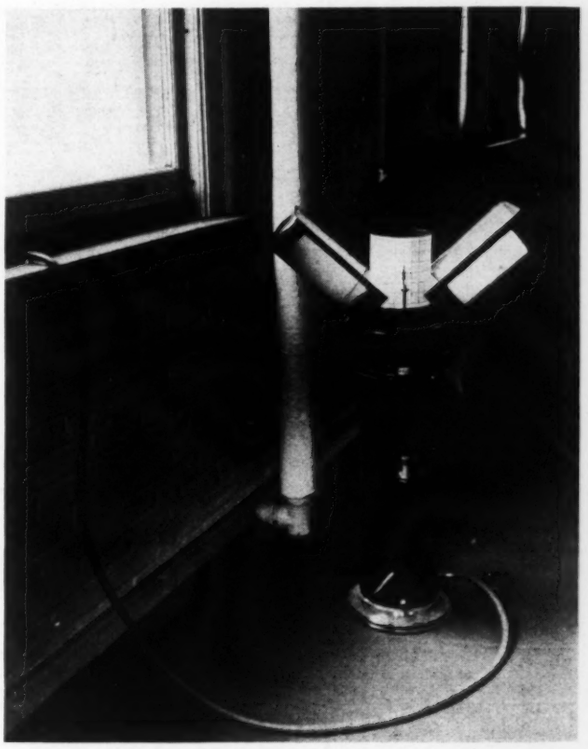

FIG. 1. RECORDING GAGE USED TO MEASURE PRESSURE DIFFERENCE ACROSS WINDOWS

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[3] For standard air density, the formula $p = 0.00048M^2$ is used to calculate pressure corresponding to a given wind velocity or vice versa, where p = pressure in inches of water and M = wind velocity in miles per hour.

[4] The Weathertightness of Rolled Section Steel Windows, A. S. H. & V. E. JOURNAL, June 1928.

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pretation of data, than to present final facts and conclusions.

Means of Obtaining Data

A record of pressure difference across two windows in one of the buildings of the University of Michigan was obtained continuously over a period of 45 days, extending from the middle of January to the first of March, 1929. A recording pressure gage with a 24-hour chart was used. A photograph of the gage, known as a hydro-recorder, is shown in Fig. 1. A narrow felt-trimmed board, with a hole through it into which a brass tube was pushed, was slipped under the lower sash, and the crack thus opened at the meeting rail was stopped with another felt-trimmed strip. The brass tube was connected to the gage which was located in the room, and the resulting record showed the excess or deficiency of the outside pressure over that inside. The connection of the gage to the window may be seen in the picture. Fig. 2 shows a part of the record during a period when there was but little wind, and Fig. 3, of a part when the wind was rather high, and from the direction of exposure. The highest momentary pressure difference recorded was 0.88 in. of water, which, in terms of wind velocity, would be about 42 miles per hour.

Two windows were chosen, on the west end of the building, one on the second floor, and one on the fourth. There was free exposure to wind from the west and southwest, as illustrated in Fig. 4. The gage was connected at the second floor window for 28 days continuously, and then moved to the fourth floor, where a record for 17 days was obtained.

Anemometer records of wind velocity and direction, as well as thermograph records of temperature were available at the University Observatory. The data of these records were transferred to the pressure difference charts as illustrated in Figs. 2 and 3. For convenient reference, the maximum pressure difference, and the average pressure difference as nearly as it could be determined, were read from the chart curve, and these quantities appear as lines 4 and 5. In this manner, all of the necessary data are synchronized on the pressure difference charts, with the graphic record of the pressure difference in direct view above.

Selecting Wind Velocity in Estimating Infiltration Loss

From Table 1, which is a summary of the wind data over approximately the 45-day period before alluded to,

Table 1—Summary Wind Data

Item	Direction from which Wind came								
	N	NE	E	SE	S	SW	W	NW	Total
1. No. of hours	97	50	129	85	57	155	179	289	1041
2. Miles of wind	569	187	892	509	308	1454	1656	2437	8012
3. Avg. velocity	5.9	3.7	6.9	6.0	5.4	9.4	9.3	8.9	7.7
4. Percent of hours	9.3	4.8	12.4	8.2	5.5	14.9	17.2	27.7	100.0
5. Percent of wind	7.1	2.3	11.1	6.5	3.8	18.1	20.7	30.4	100.0

it is seen that the average wind velocity is 7.7 miles per hour over the entire time. It is also seen that the wind is predominantly from the SW, W, and NW, nearly 70

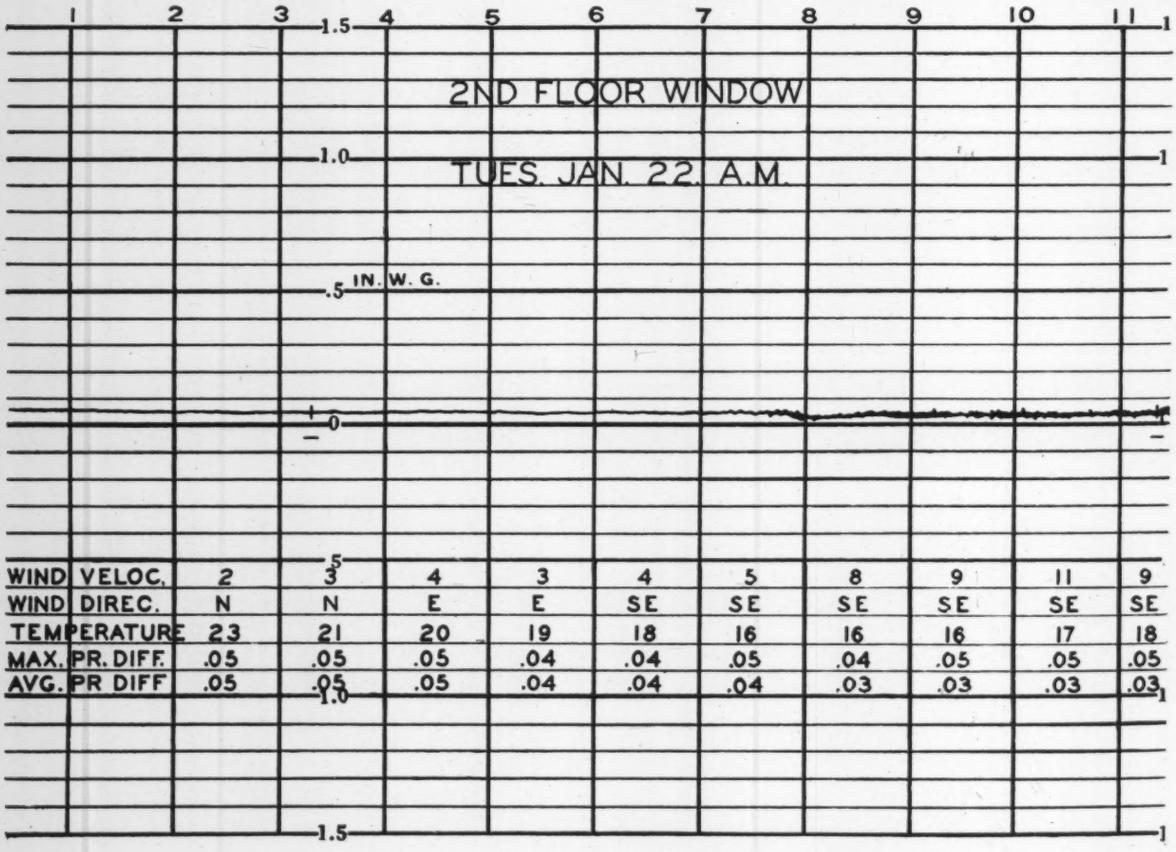


Fig. 2—Graphic Record when Wind Velocity Was Low

TABLE 2—PREVAILING AND NON-PREVAILING WINDS

WIND VELOCITY	NUMBER OF HOURS			
	PREVAILING WIND		NON-PREVAILING WIND	
	SW. W. AND NW.		N. NE. E. SE. AND S.	
	ALL HOURS OF DAY	7 A. M. TO 5 P. M.	ALL HOURS OF DAY	7 A. M. TO 5 P. M.
1. 25 M.P.H. or more.....	8	5	0	0
2. 20 M.P.H. or more.....	35	10	0	0
3. 15 M.P.H. or more.....	73	25	9	1
4. 10 M.P.H. or more.....	246	78	44	14
5. 9 M.P.H. or more.....	.....	.....	69	28

per cent of the wind coming from these three directions, and only 30 per cent from the other five.

In order to get at usable maximum values of wind velocity, Table 2 has been prepared, in which the wind is classified as *prevailing* and *non-prevailing,* and the number of hours is shown during which the wind blew at each velocity designated. Separate consideration is given to those hours of the day included between 7 a. m. and 5 p. m., which may be regarded as the ordinary period of occupancy of a building.

From the prevailing directions, the wind had a velocity of 15 miles per hour or more for 73 hours during the month and a half investigated. Counting only the 10 hours per day of ordinary occupancy, the wind had a velocity of 15 miles per hour or more for 25 hours, which is somewhat less than ten twenty-fourths of 73, indicating that most of the higher wind velocities occur at night. If it be assumed that the wind statistics for the other month and a half of the season are like those studied, it could be said that the wind from the prevailing directions has a velocity of 15 miles per hour or more during 50 hours of the time of occupancy of a building. Perhaps this would not be an unreasonable figure upon which to base allowance of heating surface requirements, considering the fact that the periods of high wind velocity will not always occur simultaneously with the periods of lowest temperature. The average speed of the prevailing wind for all those hours during which the velocity is 15 miles per hour or more is found to be about 19 miles. Hence, for this particular case, it would appear that a value of 19, or say 20 miles per hour would represent a reasonable value for wind velocity to be chosen for those sides of a building exposed to the prevailing wind. To obtain more accurate results, wind data for several seasons should be studied in this manner.

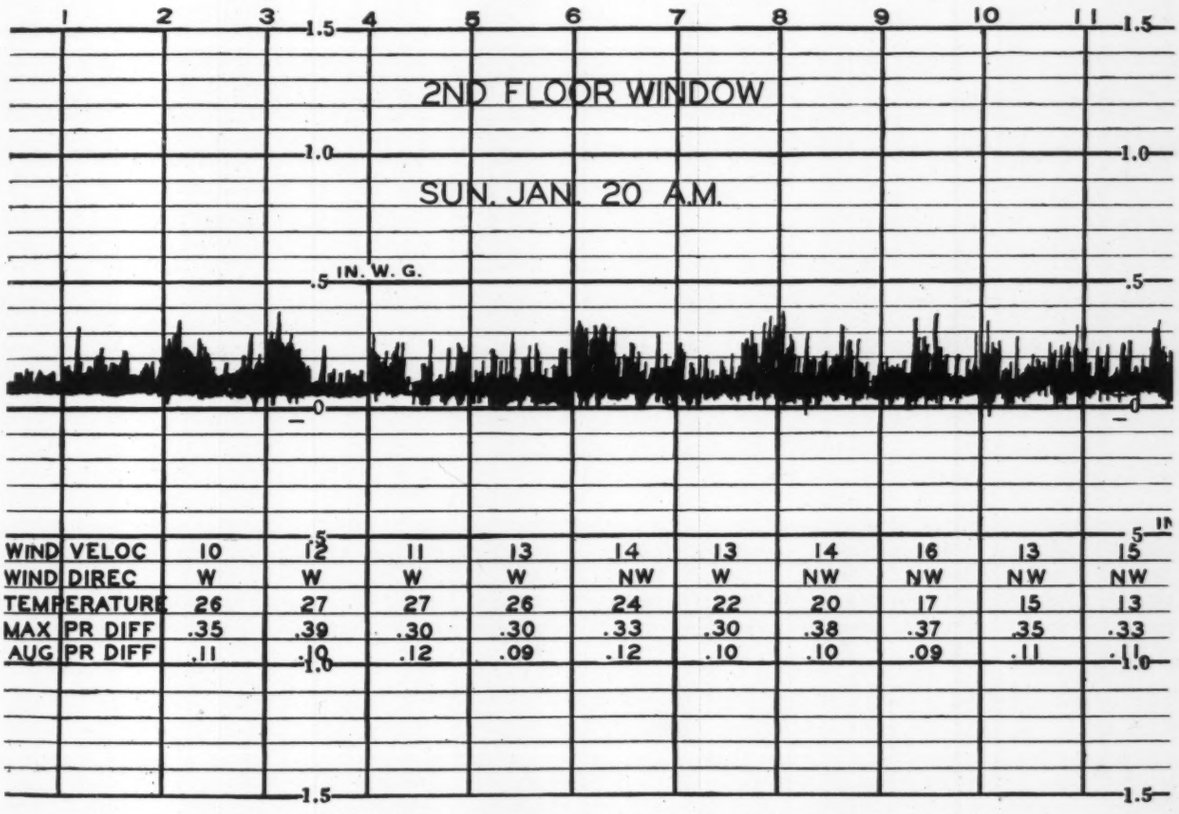


FIG. 3—RECORD OF HIGH WIND FROM DIRECTION OF EXPOSURE

Turning now to the non-prevailing winds of Table 2, it is seen that the wind had a velocity of 9 miles per hour or more during 28 hours of the time of occupancy, for the month and a half period, or perhaps 56 hours for the entire season. The average speed of the non-prevailing wind for all those hours during which the velocity is 9 miles per hour or more is found to be about 13 miles.

Having thus determined what is to be considered as the wind velocity, the next objective is to ascertain if possible what is the relation between wind velocity and actual pressure difference, and to find out if the factor

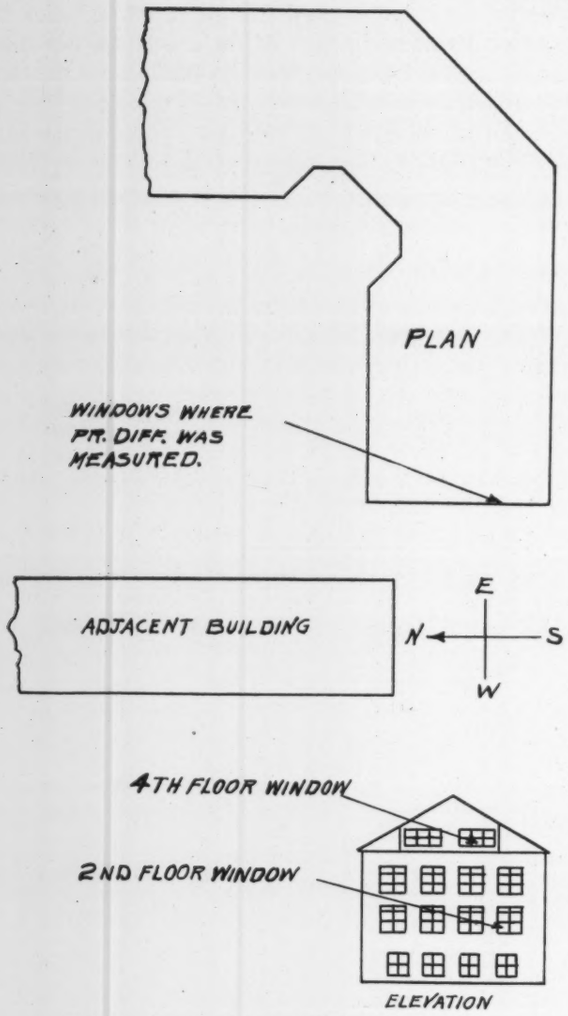


FIG. 4. SHOWING LOCATION OF WINDOWS

of 80 per cent of Part II of THE GUIDE tables is justified.

Relation of Pressure Difference to Wind Velocity

A record of pressure difference across two windows of a chosen building, one on the second floor and one on the fourth floor immediately above (see Fig. 4) was obtained in the manner heretofore described. Owing to the proximity of an adjacent building, the windows were sheltered from a northwest wind, but were fully exposed to the west and southwest. The relation between pressure difference across the second floor window and the wind velocity from the west or southwest (the direction of exposure of the window), is shown by the curve of Fig. 5. Each point represents the average of the mean hourly pressure differences for the number of hours during which the wind velocity had the particular value for which the point is plotted. The points lie quite consistently near the smooth curve, except in the region of higher wind velocity, where each one is the average of fewer values than is the case in the region of lower wind velocity.

The curve of Fig. 6 represents the case where the wind came from the northwest, which is considered separately because the window is sheltered in this direction by the adjacent building as shown by Fig. 4. The curve of Fig. 7 shows how the pressure difference on this second floor window is influenced by wind coming from directions other than southwest, west or northwest. The curves of Figs. 8, 9, and 10 present the same kind of picture of the relation between pressure differences and wind velocity for the fourth floor window.

It might have been expected, with winds coming over or around the building, from directions other than southwest, west or northwest, that there would have been a tendency to form a slight vacuum at the leeward end in which the windows were located, and thus materially reduce the excess of outside pressure over inside pressure, or even reverse the direction of excess at the upper window. This same effect might even be expected in some degree when the wind came from the northwest, blowing over or around the adjacent building (see Fig. 4). The curves of Figs. 6, 7, 9, and 10, do indeed suggest this very thing, as all of them show a slight drop after the wind attains a velocity of about 8 miles per hour. None of the curves reveal an excess of pressure inside, over that outside at the upper window, as would be anticipated by the theory of effect of temperature difference. However, there is a reason to account for this situation, which will be explained.

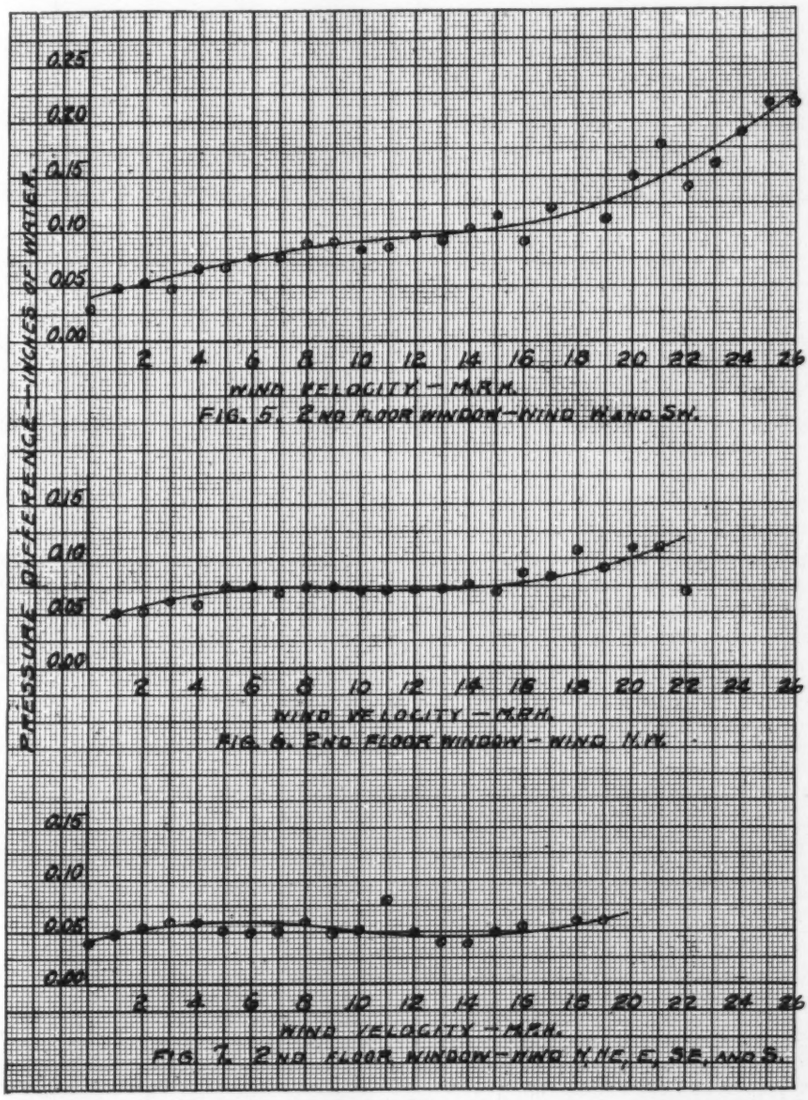
Since the principal object of this part of the study is to examine the pressure difference over windows exposed to the wind, attention will now be directed only to the curves of Figs. 5 and 8, and these may be considered to represent what would occur at any window about the building when the unobstructed wind blew against it. The pressure differences at the windows at the two levels investigated are compared in Fig. 11, where *A* is the curve of Fig. 5 (second floor window), and *B* is the curve of Fig. 8 (fourth floor window). Curve *C* of Fig. 11 is the computed wind pressure drawn in for reference. It is seen that the curves for the two windows are more or less parallel, *A* being displaced above *B* by an average amount of 0.053 in. of water. Using the average temperatures that prevailed and the height between the windows, which is about 28 ft., the head caused by temperature difference is calculated to be 0.041 in. of water, which should express the amount by which the pressure difference at the lower window is greater than that at the upper, and pretty well accounts for the actual difference of 0.053 in. between the curves. If it were not for the upward bulge in the curve *A* between wind velocities of zero and about 14 miles per hour, for which no reason seems to be evident, it is probable that the difference between the curves *A* and *B* would be almost exactly accounted for by the effect of temperature.

It would be anticipated that with little or no wind, the temperature difference would produce a negative effect at the upper window—that is, the inside pressure would be greater than the outside, and outflow would occur there. This would be evidenced by the curve *B* of Fig. 11 coming below the base line at low wind velocities, which it does not do. It is believed that the presence of gravity flues in the rooms in which the windows are located accounts for the apparent departure from theory. The gravity flue on the second floor drew air from that room, and thus tended to increase the excess of outside pressure. At the same time the loss of air here reduced the amount that would have otherwise naturally as-

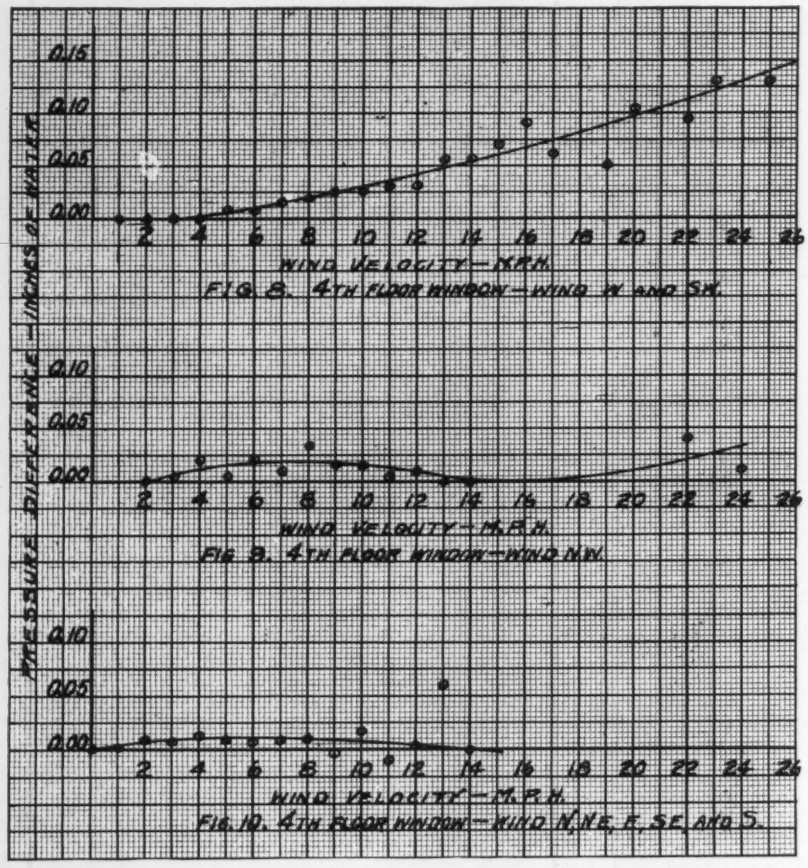
cended to the fourth floor by way of open stairs in other parts of the building, and so relieved the interior pressure on the fourth floor windows. Also the gravity flues in the fourth floor room provided an escape for air which further relieved the inside pressure there, with the net result that there was no excess of inside pressure over that outside on the fourth floor windows. It is probably merely accidental that the pressure difference at the fourth floor happened to be exactly zero with no wind.

In Fig. 12, curve *D* is the mean of curves *A* and *B*, of Fig. 11—that is, it represents the average of the pressure differences observed at the second and fourth floor windows. Curve *C* is again the pressure computed from the wind velocity. The ratio of the average pressure difference to the wind pressure is represented by curve *E*, and this is the relation that is the principal objective of this part of the study. Up to a velocity of 11 miles per hour, the pressure difference exceeds the wind pressure, as a result of the temperature effect. Above 11 miles per hour, the pressure difference is less than the wind pressure, and it is interesting to note that curve *E* seems to approach and become constant at about the value of one-half, in the region of high wind velocities. This is just what would be expected to happen in a building, when the wind pressure becomes so great as to submerge the temperature effect, and where the total leakage area for inflow on the windward faces about equals the total area for outflow on the leeward faces. The inside pressure would have to build up automatically to a value equal to half the wind pressure, so that the head available to force the air out would be equal to the head available to force an equal amount of air in.

Having chosen a value of 20 miles per hour for wind velocity as a basis upon which to determine infiltration in the calculation of heating surface requirements for rooms on exposed sides of the building, curve *E* of Fig. 12 would indicate that the actual pressure difference across the windows is about 0.55 of the pressure computed from this wind velocity, which means that the leakage is about 75 per cent of the amount corresponding to the pressure computed from the wind velocity, since leakage quantity is approximately proportional to the square root of the pressure difference. In other words, the factor to be applied in Part II of the tables of The Guide 1929, to allow for building up of inside



Figs. 5, 6, 7—Curves Showing Relation Between Pressure Difference and Wind Velocity for 2nd Floor Windows


0.15
0.10
0.05
0.00
2 4 6 8 10 12 14 16 18 20 22 24 26
WIND VELOCITY—M.P.H.
FIG. 8. 4TH FLOOR WINDOW—WIND W AND SW.
0.10
0.05
0.00
2 4 6 8 10 12 14 16 18 20 22 24 26
WIND VELOCITY—M.P.H.
FIG. 9. 4TH FLOOR WINDOW—WIND NW.
0.10
0.05
0.00
2 4 6 8 10 12 14 16 18 20 22 24 26
WIND VELOCITY—M.P.H.
FIG. 10. 4TH FLOOR WINDOW—WIND N, NE, E, SE, AND S.

FIGS. 8, 9, 10—CURVES SHOWING RELATION BETWEEN PRESSURE DIFFERENCE AND WIND VELOCITY FOR 4TH FLOOR WINDOWS

pressure should be about 75 per cent in this case.

It will be recalled that the usable maximum wind velocity for sides of the building exposed to the non-prevailing winds was chosen as 13 miles per hour. This is near the value of 15 miles, which is probably the intent of THE GUIDE to suggest for use in connection with the less severe exposures. Referring to Fig. 12, it is seen that for 15 miles per hour, the ratio of actual average pressure difference at the two windows, to the calculated wind pressure, is 0.67, which means that the leakage is about 82 per cent of the amount corresponding to the pressure computed from the wind velocity.

From this study, it appears that the factor of 80 per cent, as applied in Part II of the tables of THE GUIDE 1929, is well chosen, probably erring, if at all, on the side of being too large. However, it is indicated that the practice of allowing an additional 15 per cent in leakage loss for the sides of a building exposed to the prevailing wind may not provide an adequate margin of heating surface on those sides. For example, if 15 miles per hour be taken as the wind velocity upon which to figure infiltration on the less exposed sides, the leakage per foot of sash crack of a double hung weatherstripped wood window is 11.7 cu. ft. per hour. It has been shown that a reasonable value to be taken for wind velocity from prevailing directions is 20 miles per hour, for which the leakage is 22.9. Thus in the case studied the leakage of windows under the more extreme conditions on the more exposed sides, is nearly twice as much as that on the less exposed sides, instead of only 15 per cent greater. If this is true of infiltration loss, it is also true of transmission loss, although probably in a different ratio. In other words, in the determination of heating surface to take care of the most severe situations, it is probable that there should be a greater differential than 15 per cent in the allowances for infiltration and transmission loss between the windward sides of a building and the sides of lesser exposure.

Distribution of Leakage at Windows at Different Levels

Referring to Fig. 11, it is seen that the pressure difference at the second floor window is considerably greater than it is for the fourth floor, which, as previously explained, is practically accounted for by the temperature difference effect. Converting the pressure difference at 20 miles per hour into corresponding wind velocity, and then going to THE GUIDE table, it is found that the leakage at the second floor for a double hung weatherstripped wood window is about 17.5 cu. ft. per hour per foot of sash crack, and that at the fourth floor about 12.8 cu. ft., which calls for 0.025 sq. ft. more radiation per foot of crack on the second floor than on the fourth. With eight windows of say 50 ft. of sash crack each, including transoms, the total difference in radiation would amount to 10.0 sq. ft. Considering windows of poorer construction, and taking frame leakage into account, the total difference in radiation between the floors for rooms at the one end of the building (Fig. 4) might easily run to 40 or 50 sq. ft. The difference in heating surface requirements between lower and upper floors is thus seen to be of considerable amount even in the case of

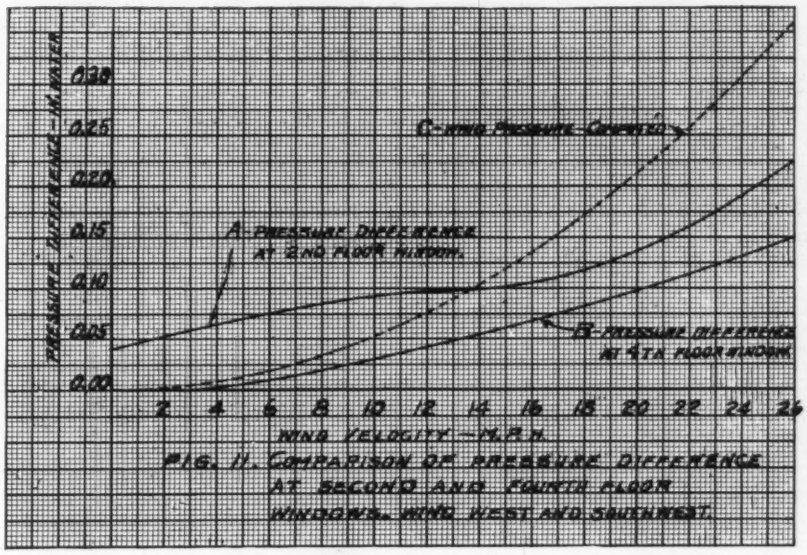

C-WIND PRESSURE COMPUTED.
A-PRESSURE DIFFERENCE AT 2ND FLOOR WINDOW.
B-PRESSURE DIFFERENCE AT 4TH FLOOR WINDOW.
0.30
0.25
0.20
0.15
0.10
0.05
0.00
2 4 6 8 10 12 14 16 18 20 22 24 26
WIND VELOCITY-M.P.H.
FIG. 11. COMPARISON OF PRESSURE DIFFERENCE AT SECOND AND FOURTH FLOOR WINDOWS-WIND WEST AND SOUTHWEST.

FIG. 11 — COMPARISON OF PRESSURE DIFFERENCE AT 2ND AND 4TH FLOOR WINDOWS — WIND W. AND S. W.

three or four story buildings. It would of course be greater still in taller buildings, although it is reduced when there is not free communication between stories by way of open staircases or other means.

In connection with the matter of extra allowance in square feet of radiation to take care of maximum infiltration, it is to be emphasized that this is not a measure of heat loss by infiltration, but is rather an allowance in capacity to meet an extreme or emergency situation. Infiltration goes on to some extent continually, so long as there is either a temperature difference or a wind, but not at the maximum rate. The column of the tables in THE GUIDE 1929 headed *Heat Loss* is likely to be misinterpreted to mean a continual drain from the heating system at the B.t.u. rate per degree temperature difference indicated, which is not the case.

In conclusion, it may be said that the study described and developed in this paper throws some light upon the relation of actual pressure difference across windows to wind pressure, for one particular case. It is indicated that something definite may be derived from a study of this kind. It is further plainly shown that temperature effect, even in buildings of moderate height, may be a factor of sufficient importance to require a different allotment of heating capacity as between lower and upper floors. A study of Weather Bureau records and a general agreement as to the interpretation of data therefrom are highly important in order to better define what shall be considered as the wind velocity upon which to determine window leakage.

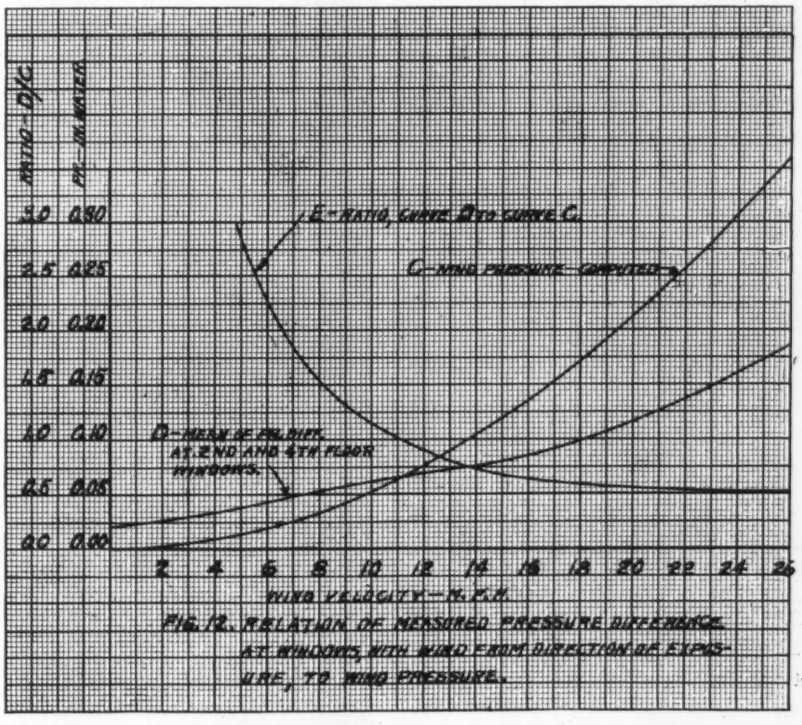

RATIO-R/C
R-IN INCHES
3.0 0.30
2.5 0.25
2.0 0.20
1.5 0.15
1.0 0.10
0.5 0.05
0.0 0.00
R-RATIO, CURVE BTO CURVE C.
C-WIND PRESSURE-COMPUTED.
O-MEAN OF PRESSURE DIFF. AT 2ND AND 4TH FLOOR WINDOWS.
2 4 6 8 10 12 14 16 18 20 22 24 26
WIND VELOCITY-M.P.H.
FIG. 12. RELATION OF MEASURED PRESSURE DIFFERENCE AT WINDOWS, WITH WIND FROM DIRECTION OF EXPOSURE, TO WIND PRESSURE.

FIG. 12 — RELATION OF MEASURED PRESSURE DIFFERENCE AT WINDOWS, WITH WIND FROM DIRECTION OF EXPOSURE, TO WIND PRESSURE

NOMINATIONS FOR OFFICERS OF THE SOCIETY FOR 1930

Your Committee appointed to prepare nominations for Officers of the Society for the coming year, 1930, takes pleasure in submitting the following list of nominees:

For President:

L. A. HARDING, 1335 Main St., Buffalo, N. Y.

For First Vice-President:

W. H. CARRIER, Newark, N. J.

For Second Vice-President:

F. B. ROWLEY, University of Minnesota, Minneapolis, Minn.

For Treasurer:

C. W. FARRAR, Buffalo, N. Y.

For Members of the Council:

Three-Year Term

R. H. CARPENTER, New York, N. Y.
JOHN D. CASSELL, Philadelphia, Pa.
ALFRED KELLOGG, Boston, Mass.
JOHN HOWATT, Chicago, Ill.

Respectfully submitted,

NOMINATING COMMITTEE

FREDERICK D. MENSING, Chairman

C. E. BRONSON
R. C. BOLSINGER
H. B. HEDGES
H. M. NOBIS
JAY R. McCOLL
O. K. DYER
EDWIN C. EVANS, Sec'y.

In accordance with the provisions of the Amended Constitution, as given, ballots containing the names of the above candidates will soon be sent to the membership for voting upon prior to the Annual Meeting in January.

Art. 9—*Section* 2. The Nominating Committee shall consist of a member designated by each Chapter, which shall effect its own organization and elect is own Chairman.

Section 3. The Committee may meet at the call of the Chairman, for discussion, but the official meeting shall be held at the Semi-Annual Meeting of the Society, at which time the candidates for each of the offices of the Society to be filled at the next Annual Meeting shall be selected by a majority vote of the members of the Committee present and voting. Five shall constitute a quorum.

Section 4. The Committee shall first secure the consent of all candidates selected, and shall if possible announce the names of the candidates at the Semi-Annual Meeting, but in any case the names of the candidates shall be certified to the Secretary of the Society at least four months before the next Annual Meeting. The Secretary shall publish these names in the October issue of THE JOURNAL.

Art. 10.—*Section* 3. The Secretary shall prepare ballots with the names of all candidates and forward them to the members, at least thirty (30) days before the date of the Annual Meeting.

Nominations for Members of Committee on Research

Although committees of the Society are usually appointed, in view of the great importance of the Committee on Research and the financial responsibility it would be called upon to assume, it has been made more representative of the entire membership of the Society by a process of *election*. The election is governed by the By-Laws for the election of officers, with the single exception, that Members of the Committee on Research are nominated by the Council instead of by a Nominating Committee. In accordance with the Regulations for the government of the Research Laboratory, adopted at the January, 1919, Meeting, the Council announces the nomination of the following members of the Committee for election to succeed those members whose present terms expire January 1930:

C. V. HAYNES, Philadelphia
W. T. JONES, Boston
J. F. McINTIRE, Detroit
F. N. SPELLER, Pittsburgh
A. C. WILLARD, Urbana, Ill.

The regulations governing the nomination and election of members of the Committee on Research are as follows:

2. ELECTION.

a. That the Council shall nominate previous to October 1 of each year, five members to fill vacancies of those retiring at the next Annual Meeting.

b. That the nominations made by Council shall be published in the October JOURNAL of the Society.

c. That prior to December 1 of any year, any ten members, over their own signatures, may nominate one or more additional members for the Research Committee, and such additional nominations shall be placed on the ballot opposite such of the nominations made by the Council as designated by those making the nomination.

d. The election shall otherwise conform to the same regulations as are provided for the election of officers.

e. Vacancies may be filled by the Council, such persons, chosen by the Council, to serve until a successor can be elected at the next annual election.

Some Studies of Airation of Garages

W. C. Randall[1] (Member), and L. W. Leonhard[2] (Non-Member), Detroit, Mich.

THE problem of heating rather than that of airation or ventilation is the main reason for the acute condition that at times prevails in garages due to the presence of carbon monoxide. This is evident from the investigation which showed that objections to carbon monoxide gas only occur, with very few exceptions, during certain periods of the cold months of the year. During the warm months of the year, practically no trouble is experienced since doors and windows are usually open. Surveys made it apparent that, to an unusual extent, adequate air can be supplied by natural means. As cold weather sets in, more and more of the ventilation openings are closed in order that comfortable temperatures can be maintained within the buildings until, on extremely cold days, very few windows are open, with a result that the carbon monoxide concentration runs high. This condition is particularly troublesome towards the end of the day, possibly due not only to the higher concentrations, but also a decreasing resistance to the effects of CO. The closing of ventilation openings usually can be charged to either an insufficient heating installation or one operated much below capacity for economic reasons.

This investigation was made to determine the use made of airation or natural ventilation as a means of diluting the carbon monoxide concentration of the air in various types of garages.

The research was conducted in four types of garages—Multi-story ramp, Auto service garage, Alley repair shop and the Private home garage, and extended over portions of three years, during both summer and winter conditions. It consisted of a study of the airation, the carbon monoxide concentration of the air within the buildings, and at times the effect of carbon monoxide poisoning upon the employes. In the presentation, each type of garage will be treated separately:

Method

Most of the surveys were conducted in the following manner:—An airation survey of the building was made to determine the quantity of air entering and leaving. This was done by multiplying the area of each opening by the average velocity of air passing through that opening as measured by an anemometer. With such a method, any important discrepancies were easily found because the total inflow of air should approximately equal the total outflow. Samples of the air within the building were taken in at least six localities on each floor, at the floor level, and in the later surveys also at a level about 5 ft. above the floor. The samples were taken by purging the original air from 250 cu. cent. bottles by means of a rubber aspirator bulb with attached scrubber, and were analyzed by the Pyrotannic Acid Method[3] for quantitative determination of carbon monoxide in air. Employes were questioned to determine whether or not they experienced any symptoms of carbon monoxide poisoning.

Multi-Story Ramp Garages

This type of garage has become popular in congested districts of all large cities for the protected parking and storage of automobiles. It enables the storage or parking of a large number of cars over a relatively small ground area and differs from the ordinary service garage by having a large percentage of the floor space available for parking. Little or no repair work is carried on within the building.

Three such garages were studied: Two were five and eight stories high, respectively, having three walls pierced with openings and the maximum total square feet of openings on each floor are less than 3 per cent of the floor area, with the exception of the first floor where there are large door openings. Natural ventilation is employed throughout with the exception of a small fan exhauster, located near the roof on the uppermost floor where cars are washed and the windows are kept closed most of time to maintain comfortable working temperatures.

The buildings are heated by direct steam radiation and a temperature of not less than 40 deg. fahr. is maintained.

A larger garage of this same type was also studied. It was a 12 story structure, Fig. 1, about 150 ft. x 100 ft. with a cubical content of about 1,500,000 cu. ft. It has steel windows on all four elevations and is similar to the others in design and layout. It differed as to the heating system and employed unit heaters instead of radiators. The temperature in this building was maintained between 50 and 60 deg. fahr.

From the garage records, it was found that the average total number of cars handled for one day during the busiest season of the year was about 460 for the five-story garage, 1,050 for the eight-story, and about 1,450 for the 12-story garage. These figures are based upon one day's average of the total number of cars handled during one week not including Sunday. Of the total number of cars handled during any day but Sunday, about 60 per cent enter the buildings between the hours of 8:00 a. m. and 9:30 a. m., and leave between 3:30 p. m. and 5:00 p. m. During these periods, which might be termed the daily rush hours, the CO. concentrations varied from one-half of one part to two parts per 10,000 of air, and such high concentrations would not have

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For presentation at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January 1930.

[3] Technical Paper No. 373, Department of Commerce—The Pyrotannic Acid Method for Quantitative Determination of Carbon Monoxide in Blood and in Air—by R. R. Sayers and W. P. Yant.

FIG. 1—THE TWELVE-STORY RAMP GARAGE

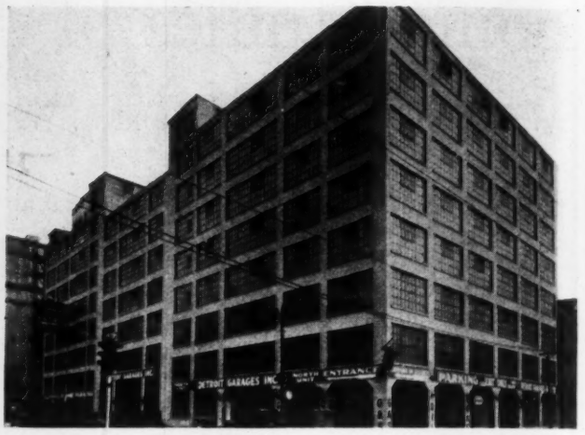


FIG. 2—EIGHT-STORY RAMP GARAGE

existed if more windows had been opened, allowing more air to enter and leave the building, and more heat had been supplied to maintain the desired temperature.

One investigation on the 12 story garage showed that about 1,000 cars moved in or out of the garage during the rush period of two hours. The average time interval for parking the car or from the starting of the motor until the car leaves the building is about 2 min. The carbon monoxide exhausted was probably 1½[4] cu. ft. per car minute. This would mean that about 3,000 cu. ft. of CO. was exhausted into the garage during the two hours, or an average of about 25 cu. ft. per minute.

The airation survey showed that about 25 per cent of the window openings or the equivalent of less than 1 per cent of the floor area were being utilized or about 50 sq. ft. for inflow and about the same for outflow per floor. The average velocity of the air passing through the window openings as measured by an anemometer was about 250 ft. per minute. The total inflow and outflow was approximately 150,000 cu. ft. per minute or six air changes per hour. This shows a concentration of about 1 part CO. in 6,000 of air or about 1½ parts per 10,000. As soon as the rush period was over the CO. was soon diluted to about ½ part per 10,000 or less.

Ordinarily,[5] exposure to concentrations as high as two parts per 10,000 for such a short time would show little, if any, symptoms of CO. poisoning. It was found that such a high concentration at the close of a day's work was sometimes just enough to increase the CO. concentration in the blood of the employes, who had been exposed to low, ineffective CO. concentration all day, to the extent that severe headache resulted.

The suggestion was made that more windows be opened during the rush hours in the afternoon, but this was met with the answer that the inside temperature would be lowered so much that the patrons would protest, thinking their cars had been exposed to such a temperature all day, thus defeating one of the purposes of protected parking. More heat during the rush hours would, undoubtedly, have eliminated the objectionable situation, since more effective airation would have been possible.

Although the 12 story garage, Fig. 1, handled considerably more cars than either of the other two, Figs. 2 and 3, the CO. concentrations were practically the same, since the cubical contents per car in all these garages was not materially different.

The carbon monoxide seemed to be well diffused or distributed. Table 1 shows there was practically no difference in the concentration of carbon monoxide at the floor level and five feet above the floor. This is probably due to the constant agitation of the air caused by the cars being driven to and from the parking stalls. The concentration on the lower floors was always less than on the upper floors due to the large volume of air admitted through the large entrance and exit doors which were being opened and closed continuously. No pockets of high concentration were found.

A typical survey for the summer conditions in one

TABLE 1—AIRATION SURVEY OF MULTI-STORY RAMP GARAGE

AVERAGE CONDITIONS WEST WIND - 6 MILES PER HOUR INSIDE TEMPERATURE - 60° OUTSIDE TEMPERATURE - 45°				SURVEY A	
FLOOR NUMBER	AVERAGE CO IN PARTS PER 10,000 OF AIR		AVER. VELOCITY OF INFLOW FT. PER MIN.	WINDOW INLET OPENINGS SQ FT.	INFLOW VOLUME C.F.M.
	AT FLOOR	5 FT. ABOVE FLOOR.			
12	0.4	0.4	235	55	12,900
11	0.8	0.8	250	30	7,500
10	0.4	0.4	240	58	13,900
9	0.4	0.4	250	47	11,700
8	0.4	0.4	250	49	12,200
7	0.4	0.4	260	45	11,700
6	0.4	0.4	280	43	12,000
5	0.4	0.4	270	47	12,700
4	0.8	0.8	265	61	16,200
3	0.4	0.4	250	52	13,000
2	TRACE	TRACE	240	55	13,200
1	TRACE	TRACE	225	80	18,000
TOTAL	5.0	5.0		622	155,000
AVERAGE	0.41	0.41	251	51	12,900

[4] Transactions of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1921, p. 311.

[5] Physiological Effects of Automobile Exhaust Gas and Standards of Ventilation for Brief Exposures—Yandel Henderson, H. W. Hoggard, M. C. Teague, A. L. Prince and Ruth M. Wunderlich. *Journal of Industrial Hygiene*, Vol. 3, July 1921.

FIG. 3—TYPICAL FIVE-STORY RAMP GARAGE

FIG. 4—THE TWO-STORY AUTO SERVICE GARAGE

of the five story ramp garages is shown in Table 2.

It will be noted that but a trace of carbon monoxide was found at any time. The number of cars handled in and out was about 200 for the four hours of the survey. The CO. exhausted was probably not more than 1 cu. ft. per minute and based on an average of 1½ min. running per car, this would mean about 300 cu. ft. of CO. or 75 cu. ft. per hour, somewhat less than 1½ cu. ft. per minute—with approximately 60,000 cu. ft. of air per minute blowing through the building, this would mean about 0.25 parts per 10,000 of air, which is very hard to measure accurately and is indicated in the table as a trace.

Auto Service Garage

The modern auto service type of garage, Fig. 4, is usually a single or double story structure in which automobiles are both stored and repaired. In the case of the two story garage, the cars are usually stored on the lower floor, while the repair work is normally done on the second floor. The two floors are connected by a ramp or elevator for communication.

In the course of repair work, particularly light repairs and service, the engines exhaust directly into the working zone and contaminate the surrounding air with carbon monoxide gas. During these operations, the car is seldom in motion and the exhaust gases are not so readily diffused as in the Ramp Parking garage.

TABLE 2—AIRATION SURVEY OF FIVE-STORY RAMP GARAGE

AVERAGE CONDITIONS S.W. WIND - 6 MILES PER HOUR			SURVEY B		
TIME A.M.	FLOOR NUMBER	AVER. CO IN PARTS PER 10,000 OF AIR	AVER. VELOCITY OF INFLOW FT. PER MIN.	WINDOW INLET OPENINGS SQ. FT.	INFLOW VOLUME C.F.M.
9:00	5	TRACE	307	46	14,000
10:00	4	TRACE	328	50	16,500
11:00	3	TRACE	333	40	13,300
12:00	2	TRACE	305	52	16,000
				188	59,800
AVERAGE PER FLOOR		TRACE	318	47	14,900.

Observations and tests were made in three garages of this type, one being a two story structure, Fig. 4, which employed both natural and mechanical means of getting air into and out of the building. Mechanically, air was drawn in through individual ducts, forced through unit heaters and exhausted at about 500 ft. per minute velocity near the ceiling. These heaters were provided with a damper which could be adjusted to provide total or partial recirculation of inside air. A mechanical exhaust duct system in the floor exhausted about 50 per cent of the total inlet capacity of unit heaters while open windows provided the remaining necessary exhaust outlets, and, at the same time, additional inlets.

This building, Fig. 4, has a content of about 500,000 cu. ft. Steel windows are used in all four elevations. The total ventilating window area available is about 2,000 sq. ft. or about 4 per cent of the floor area.

Tests were conducted in this garage during the months of January and February, a time when repair work was not at a peak, although the average amount was being done for that particular time of year. Most of the tests were made with the unit heaters set for total recirculation just as they had been operating probably ever since installation and, as a result, a great many windows were found open to provide inlet as well as outlet of air.

One survey showed that about 35 cars moved in and out during the day for all types of repairs from minor adjustments to major repairs. As nearly as could be observed, the average time that the engine was running was about eight minutes. This would mean about 280 total car minutes, and based on 1½ cu. ft. of CO. being exhausted per minute per car, there would be exhausted a total of 420 cu. ft. of CO. There were about 95 sq. ft. of windows serving as inlets for air, and a similar amount for outflow, or a total of approximately 0.5 per cent of the floor area. The average inflow velocity was about 180 ft. per minute, giving an inflow and outflow of about 18,000

cu. ft. per minute or about 8,000,000 cu. ft. per eight hour day, two air changes per hour. This volume also includes the inflow through the entrance door. Under such conditions, the carbon monoxide concentration ranged from a trace to about one-half of one part per 10,000 of air, which, of course, registered no ill effects upon the workmen. On this particular occasion, the inside temperature was about 60 deg. and the outside temperature averaged 25 deg., survey B—Table 3. The unit heaters were operated on a 100 per cent recirculating basis.

During a rush period when 50 or more cars are handled each day, and the same volume of air flowing through the building, the average carbon monoxide concentration would be about one part per 10,000 and at times possibly two parts per 10,000 or more. Under such conditions, if the weather was cold, and less windows were open the conditions would be acute. This is particularly true when the men have been exposed to, say, one-half part of CO. per 10,000 of air during most of the day and, due to an increase in the number of cars moved near the end of the day, the CO. concentration is sometimes raised to two parts or more per 10,000 for a short time. The answer to this is that in accordance with the volume of CO. exhausted from running autos, sufficient air must be introduced to dilute the CO. concentration below a disagreeable degree and whether this required air is introduced through open windows, or otherwise, it must be heated to a comfortable temperature.

The foregoing statements are made on the basis that no direct method of exhausting the fumes from the cars is used to take the gases through some exhaust duct system to the outside.

Tests were made on another day under almost identical conditions, with the exception that no air was recirculated through the heaters. Under these conditions, it was found that about 20,000 cu. ft. of air per minute was passing through the building and the carbon monoxide concentration was the same as before—ranging from a trace to about one-half of one part per 10,000 of air. The outside temperature was 17 deg. and the inside temperature was about 56 deg., survey C—Table 3.

Table 3—Results of Two Airation Surveys of a Two-story Auto Service Type Garage

AVERAGE CONDITIONS-SURVEY B WEST WIND-4 MILES PER HOUR INSIDE TEMPERATURE-62° OUTSIDE TEMPERATURE-25°				AVERAGE CONDITIONS-SURVEY C NORTH-WEST WIND-6 MILES PER HOUR INSIDE TEMPERATURE-56° OUTSIDE TEMPERATURE-17°		
SURVEY NUMBER	FLOOR NUMBER	AVER. CO IN PARTS PER 10,000 OF AIR		AVER. VELOCITY OF INFLOW FT. PER MIN.	INLET OPENINGS SQ. FT.	INFLOW VOLUME C.F.M.
		AT FLOOR	5 FT. ABOVE FLOOR			
* B	1	0.4	0.4	180	95	18,000
	2	0.5	0.5			
** C	1	0.5	0.5	500	33-U.H 20-W	20,000
	2	0.5	0.5	150		

* DURING SURVEY B AIR WAS INTRODUCED THRU WINDOWS ONLY.
** DURING SURVEY C AIR WAS INTRODUCED THRU UNIT HEATERS AND WINDOW OPENINGS. THE 500 VELOCITY WAS THRU THE HEATERS AND THE 150 WAS THE VELOCITY THRU 20 SQ FT. OF WINDOW OPENING.

In either case, only a portion of the total available ventilation was being used, but if more windows had been opened under the conditions, without supplying more heat, the inside temperature would, undoubtedly, have been lowered to an uncomfortable degree. This was proven out by other tests made on days when the outside temperature was very close to zero, under which conditions the inside temperature varied between 48 and 52 deg. Such temperatures were uncomfortable, but the workmen were content to sacrifice comfort for air low in CO. concentration.

In general, it can be said that the carbon monoxide was well diffused; that is, the concentration at the floor level and 5 ft. above the floor was practically the same, and this seemed to be true whether the air was admitted through the window openings as natural ventilation or through the unit heaters as mechanical ventilation. Probably it was the agitation of air caused by the high velocity of the unit heater exhausts, in either case, which caused such a complete diffusion of the carbon monoxide gas.

This garage was cited as one in which considerable trouble was experienced from high concentrations of carbon monoxide gas but results of tests conducted on 14 different days, when conditions were most favorable for high CO. concentrations, seemed to indicate that there was more discomfort experienced from inadequate heating than from carbon monoxide poisoning.

Two other garages of the auto service type were studied, but not so extensively as the one just discussed—These are of the common single story type, having three walls pierced with ventilation openings whose total area is about 3 per cent of the floor area. Natural ventilation is employed throughout, and heat is supplied by direct steam radiation.

As a result of the few tests conducted in these buildings, and information gathered from the management, it was found that as long as the outside temperatures permitted the opening of at least 50 per cent of the total window ventilators, practically no symptoms of carbon monoxide poisoning were experienced.

Tests made during near zero weather showed that the CO. was not very uniformly diffused throughout the building. Air samples taken at the floor level showed slightly higher concentrations than those taken about 5 ft. above the floor, and the highest concentration recorded was between 1½ and 2 parts per 10,000 of air. This condition existed directly after a motor tuning operation, but only for a few minutes until the doors were opened to clear the building of the smoke and gas, or opened to let a car in or out. During such cold weather the windows and doors are usually kept closed, but as soon as a motor is run for tuning or other adjustments, the doors are opened until the operation is completed. In other words, the workmen have learned their lesson; they prefer to dilute the poisonous gas before it has time to register any ill effects upon

them, but in doing so, they usually sacrifice comfortable working temperatures, at least temporarily.

Alley Repair Shops

Garages of this type are small auto repair shops usually found up an alley and located in an old barn or storage shed or some other structure which was never designed to serve such a purpose. Usually these shops can only accommodate from four to six cars and are provided with no other ventilation openings except the service doors and possibly a window or two.

A few of the shops studied are heated by direct steam radiation, others by stoves and a few are not provided with heating equipment of any kind. Where no heating equipment is provided, practically no work is done during severe cold weather.

As a result of observations made in a number of these repair shops, and of information gained from questioning employes and the proprietors, it was found that, on the average, carbon monoxide poisoning is not as serious a problem in these shops as it is in the Service Garages. The reason for this is that a motor is seldom run within the building unless the doors are open. Practically all motor testing and running-in of bearings is done outside. It is interesting to note that one was provided with an old flexible metal hose for piping the exhaust to the outside.

Private Home Garage

A few tests were conducted in a private, two-car garage to determine the rate of increase of the carbon monoxide concentration of the air for the conditions with the engine idling and the engine racing, while all doors and windows of the garage were tightly closed.

A four cylinder Chevrolet car was placed in a double garage, and after all doors and windows were closed, the motor was started and allowed to run at an idling speed for 10 min. Samples of the air at floor level and breathing level were taken every 2 min. At the end of the 10 min. period, the motor was stopped, the doors were opened and the garage was allowed to air for about 5 min. Then the doors were again closed, the throttle on the motor was set at a racing speed and the same sampling procedure was followed as before.

On the basis that the garage contained about 3,000 cu. ft. of air, it is seen that with the engine exhausting between 1 and 1½ cu. ft. of carbon monoxide per minute, the CO. concentration would be about four to five parts per 10,000 for the first minute of running, which, of course, would be a dangerous concentration to breathe. After 15 min. running, the concentration would be between 50 and 60 parts per 10,000 which would probably be fatal. Knowing this to be the case, the survey was made by going in and out through an outside door every 2 min. while air samples were being taken. Owing probably to the fact that air was introduced during each entrance and exit, the samples were not indicative of the true condition and, since no reasonable check on the data was obtained, the results of these tests are not shown. Certainly all indications pointed to the fact that a small, closed garage is no place for a man to be if the engine is running.

Conclusions

Multistory Ramp Garage

1. Sufficient air can be furnished through window openings, as natural ventilation, to keep the carbon monoxide concentration below a harmful or even disagreeable degree.

2. Higher concentrations of carbon monoxide were found to occur on extremely cold days when it became necessary to close all or all but a few windows in order that comfortable inside temperatures might be maintained.

3. Tests showed that the carbon monoxide content of the air in this type of garage is practically the same at the floor level as it is five feet above the floor. This is due to the fact that a motor is seldom running unless the car is in motion and the gas is well diffused by the action of cars going up and down the ramps.

4. Rush periods occur between the hours 8:00 a.m. to 9:30 a.m. and 4:00 p.m. to 5:30 p.m. during which periods 60 per cent of the total cars handled throughout the day enter and leave the buildings respectively. The CO. concentrations during these periods were found to range from one-half to two parts per 10,000 of air, depending upon the amount of air movement.

5. The carbon monoxide concentration between the rush hours was almost always below one-half part per 10,000 of air which produced no ill effects.

6. The CO. concentration during the rush hour was sometimes high enough to increase the existing low concentration in the blood of the attendants to the extent that headache developed.

7. The total available ventilation area was never utilized except on hot days in the summer.

Auto Service Garage

1. Practically no trouble would be experienced from carbon monoxide poisoning in the garages studied if some reasonable percentage of the available ventilation areas were utilized.

2. Where unit heaters are employed, the carbon monoxide is more thoroughly diffused than where direct radiation is used.

3. In the garages studied, sufficient heat was not supplied in cold weather to warm the necessary volume of air to a comfortable temperature.

Alley Repair Shops

1. Carbon monoxide poisoning is not a serious problem in these shops, since motors are seldom operated within the building unless the doors are open.

Private Home Garages

1. A four cylinder motor operating at a racing speed in a two car garage with doors and windows closed would produce a concentration of carbon monoxide which would probably cause death to anyone exposed to it for 10 min. or more.

PRESIDENT'S PAGE

THE fall term has commenced and Society activities are branching out. As the Chapters open their fall programs, it will be my pleasure to visit five of the Chapters during October and one of the Society matters uppermost in mind is the desirability of increasing our membership. C. W. Farrar is in charge of the increase of membership work and will co-operate closely with the Chapter Officers in securing an adequate number of the younger men in the profession, as well as those of greater experience who should belong to this growing organization.

Mr. Farrar will appear at the various Chapter Meetings and tell about the membership work now under way.

The Council, at its recent New York meeting, announced its nominations for members of the Committee on Research.

An outstanding event in the annals of engineering is the World Engineering Congress which opens this month in Tokio, Japan and the official delegates representing the Society will be W. H. Carrier, Kunisuke Sekido and Saito Shozo.

Mr. Carrier will present a paper before this World Engineering Congress on the Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort, in which he was assisted in the preparation by a special committee appointed by the Council.

To open the new year, the event that will make history in the heating and ventilating profession is the First International Heating and Ventilating Exposition, to be held during the same week as the Annual Meeting 1930 at the Commercial Museum, Philadelphia. Already over 150 of the leading equipment manufacturers have made preparations to demonstrate their new products and in addition to the displays of modern apparatus a number of historical and educational exhibits have been arranged.

There will be thousands who will visit the Commercial Museum during the week of January 27th. Each person will take away with him something of value that will assist him in the practice of his profession. Not only will the exhibits appeal to the scientific mind but the layman will find much to attract him.

The sessions of the Society's Annual Meeting have been arranged so that ample opportunity will be given for discussion of technical papers and at the same time the comfort and convenience of members and guests have been provided for, so that they will have ample opportunity to visit the Exposition.

The Advisory Committee of the Society for the Exposition and the Cooperating Committee of executives from allied organizations have done their work well. The local arrangements committee for the Society's Annual Meeting consisting of John Cassell, *Honorary Chairman;* R. C. Bolsinger, *General Chairman of Arrangements;* Lee Nusbaum, *Finance Committee Chairman;* M. C. Gillett, *Ladies Entertainment Committee Chairman;* F. D. Mensing, *Transportation and Publicity Committee Chairman* and A. J. Nesbitt, *Banquet Committee Chairman,* anticipate that they will welcome the largest attendance at any Society meeting and have prepared a splendid program for those who visit Philadelphia during the week of January 27, 1930.

W. H. CARRIER

ELMER A. SPERRY
CHAIRMAN AMERICAN
COMMITTEE

K. SEKIDO

A. S. H. & V. E. Delegates to World Congress

WHEN the World Engineering Congress opens in Tokio, Japan, there will be a delegation of 300 engineers from America.

The Council of the Society has appointed three official delegates, W. H. Carrier, Newark, N. J., Saito Shozo, Tokio, and Kunisuke Sekido of Nakano.

At one of the sessions of the World Engineering Congress, Mr. Carrier will present a paper on the Control of Humidity and Temperature as Applied to Manufacturing Processes and Human Comfort, prepared by Mr. Carrier and a special committee which included: O. W. Armspach, Chicago; H. W. Ellis, Milwaukee; W. L. Fleisher, New York; H. P. Gant, Philadelphia; F. C. Houghten, Pittsburgh; S. R. Lewis, Chicago; C. P. Yaglou, Boston; L. A. Harding, Buffalo; F. R. Still, New York and Prof. A. C. Willard, Urbana, Ill.

Mr. Carrier is second vice-president of the Society and is president of the Carrier Engineering Corp., Newark, N. J. His professional career has been devoted to the science of air conditioning engineering and is credited with numerous inventions which have aided the progress in his particular field. His work began in 1901 on his graduation from Cornell University. He served with the Buffalo Forge Co., 1901 to 1913, after which the Carrier Engineering Corp. was formed. Mr. Carrier was born November 26, 1876 at Angola, N. Y., and after receiving his preliminary education in country schools, he taught school while preparing for more advanced courses.

Mr. Carrier has been on the Council of the Society for a number of years, has served on many of its technical committees, is a past president of the American Society of Refrigerating Engineers and a member of many other scientific organizations.

Kunisuke Sekido has the distinction of being the Society's oldest member in Japan having been elected in 1903. He received his preparatory education in Tokio schools and graduated from the Imperial University of Tokio in 1896. During his connection with Takato & Co. he served in the New York office for two years and in 1902 spent a year in the study of heating systems in England, Germany, France and Holland.

At present he is a consulting engineer at Nakano, Japan.

Saito Shozo is the owner and managing director of the Saito Shozo firm of importers, engineers, contractors and has been a member of the Society since 1923. He is a member of the Committee for the World Engineering Congress representing the *Japanese Society of Heating and Refrigerating Engineers* and is serving as a delegate not only for the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS but for the *American Society of Refrigerating Engineers* and the *Institution of Heating and Ventilating Engineers,* London, England.

In a recent letter Mr. Saito tells of the meeting of the twelve engineering organizations of Japan which have completed all arrangements in connection with the management of the Congress and classification of members.

He reports that of a total of 544 papers which have been submitted to the congress, 319 come from Japanese sources and 225 are by foreign engineers.

The American delegation of 300 headed by Elmer A. Sperry, chairman of the American Committee and also president of the *A. S. M. E.,* includes 17 official delegates from the United States government. The congress, sponsored by the Kogakkai, the engineering society of Japan, will be attended by engineers from nearly every country in the world.

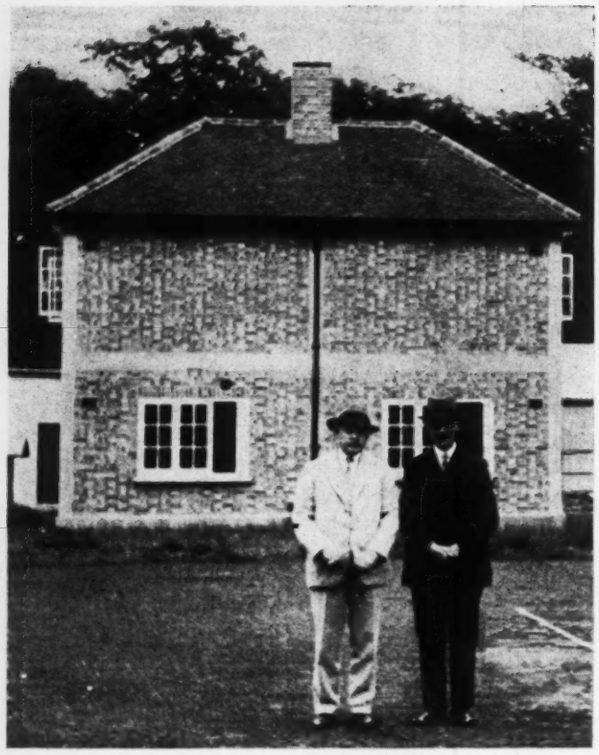
The purpose of the congress is to promote international cooperation in the study of engineering science and problems in all its branches and in stimulating a sense of brotherhood among engineers everywhere.

The American Committee of the World Engineering Congress of which President Hoover is honorary chairman has been active for a year and a half in fostering interest in the congress and arranging for American participation in its deliberations. Due to the efforts of the American Committee 75 papers have been prepared by some of the foremost American authorities in all the fields of engineering to be presented at the technical sessions of the Congress.

Among those who have been active in promoting interest in the World Engineering Congress as chairman of the various committees are: Elmer A. Sperry, chairman of the American Committee; J. W. Lieb, finance; D. C. Jackson, technical program; F. B. Jewett, transportation; O. C. Merrill, entertainment; J. H. McGraw, publicity; George W. Fuller, promotion and attendance. The following are members of the executive committee: J. W. Lieb, chairman, Gano Dunn, George W. Fuller, Maurice Holland, D. C. Jackson, F. B. Jewett, J. H. McGraw, Charles F. Scott, Elmer A. Sperry, W. E. Wickenden and Calvin W. Rice.

The American delegation will join about one hundred European engineers in San Francisco and sail for Japan on October 10. Two steamers, the President Jackson of the Dollar Line and the Korea Maru of the Nippon Yusen Kaisha have been officially designated to convey the delegates across the Pacific. By special arrangement with the steamship companies the ships will call at Honolulu and remain there for two daylight days, arriving at Tokio one day before the opening of the congress.

On October 2 a special train will take a number of the delegates from New York and vicinity to Washington where they will be received by President Hoover at the White House to be followed by a dinner and reception given to the delegates by the Japanese Ambassador, Katsuje Debuchi, it being the Ambassador's wish to send the delegates "from Japanese soil to Japanese soil." After the dinner, the party will proceed to San Francisco. A second group of engineers from the Eastern part of the country will leave New York on October 6, proceeding directly to San Francisco via Chicago, arriving on the coast just before embarkation.



THORNTON LEWIS AND J. ROGER PRESTON

British Members Entertain President Lewis

The American Society members in England, under the leadership of W. Nelson Haden of Trowbridge, entertained Pres. Thornton Lewis of the Society, at a dinner on July 18, in London, where the Officers and Council of the Institution of Heating and Ventilating Engineers of Great Britain were guests.

The following is a list of those who attended:

A. H. Barker, London.
R. C. Ching, London.
J. W. Cooling, London.
S. Fox, Halifax.
E. Herring, London.
C. Ingham Haden, Trowbridge.
E. G. Hill, Hull.
F. H. Horton, London.
E. R. Jones, Stourbridge.
Thornton Lewis, Philadelphia.
J. L. Musgrave, London.
S. Naylor, Halifax.
W. W. Nobbs, London.
A. B. Potterton, London.
J. Roger Preston, London.
J. Nelson Russell, London.
H. B. Watt, London (Secretary of *I. H. V. E.*).

Mr. Haden acted as Toastmaster, and President Lewis presented J. Roger Preston, President of the British Institute, with an engrossed resolution greeting from the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, adopted at the recent Summer Meeting held in Canada.

President Lewis in replying to a toast by Mr. Preston, told of the trend of heating and ventilating in the United States.

The recent bulletin from Mr. Preston, to members of the British Institution, stated that in large places of assemblage developments in both England and the United States are similar, but in general heating practice there is a greater tendency in the United States to use indirect and concealed radiation, while the trend in England is to utilize as much direct radiation as possible with less regard for air temperature.

On the 7th of August, Mr. Preston escorted President Lewis to the Building Research Station at Watford where Messrs. Barrett and Dufton demonstrated the work that is being done by the British Government in collaboration with the *Institution of Heating and Ventilating Engineers*. The accompanying photo shows Messrs. Lewis and Preston and the experimental house in the background.

A visit was also made to the Bureau of Hygiene and Tropical Diseases maintained by the government, where some interesting studies in health and comfort are being made.

Exposition in Philadelphia to Attract Thousands

FOR the first time in the history of heating and ventilating, every branch of the industry will be represented in an exposition devoted exclusively to thermal equipment of vital interest to heating and ventilating engineers, architects, contractors, building owners and managers, home owners and builders and allied fields.

This International Heating and Ventilating Exposition will be held in the Commercial Museum Building, Philadelphia, Pa., during the week of January 27 and at the same time as the Society's Annual Meeting, which will take place at the Benjamin Franklin Hotel.

The steady and almost phenomenal advances and improvements recently made in the industry reads like a romance. Less than a quarter of a century ago specialized heating and ventilating was in its infancy and not a recognized part of the engineering profession. Even medical men of the old school scoffed at mechanically conditioned air.

As the equipment progressed from the days of the coal stove with its humidifying pan of water, through the rapid stages of modern advancement, science became keen to its possibilities until today these many millions are represented in the design, production and installation of modern equipment so that the industry

is rated as one of the most important branches of the world's work.

Increased efficiency demands the highest type of equipment in providing heating and ventilating comfort. Effects of temperatures and humidity conditions are of vital interest in every line of production. Thermal engineering is attracting the attention of every business man, home owner, club manager, hotel man, theater owner, school superintendent and practically every other person in charge of buildings where persons assemble for business or pleasure.

This exposition in Philadelphia will not be an experiment. It will register the culmination of six years of careful planning and development, rounding itself out into one of the most complete and comprehensive expositions of an individual industry ever undertaken.

Through the concentrated effort of the men who have been the backbone of the heating and ventilating industry, the men who have brought it up to the high standard of efficiency it now enjoys, every type of equipment, apparatus, instrument, adjunct and material entering into the business will be brought together under one roof at the Commercial Museum where they may be studied by the industrialist, business man, architect, contractor and other allied interests.

Nearly 200 companies have already taken space at the exposition and, according to information coming from these corporations, their exhibits will be under the supervision of expert engineers whose training will specially fit them to explain every detail of the efficiency and economic value of their products.

These myriad exhibits will be made in the immense Museum Building all on one floor, which is cement paved. On one side of the building a driveway gives ample unloading space and on the other a private Pennsylvania Railroad siding with a ten car capacity will aid the work of exhibitors.

Water drain or gas connections are provided and electric service is available in 110 or 220 volts, alternating current in any location while D. C. 220 volts is available only in limited locations. In this way operating units will be demonstrated for the benefit of visitors.

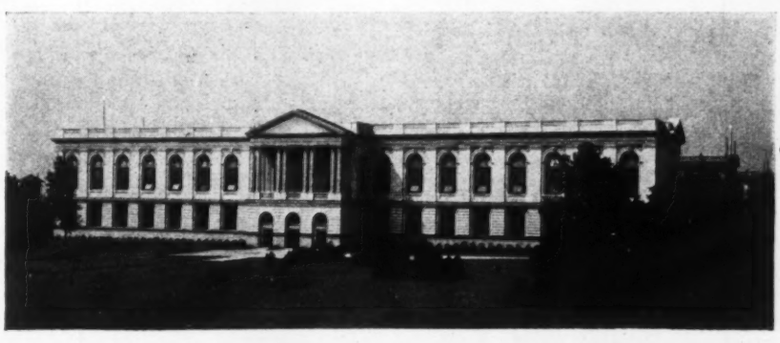
Pres. Thornton Lewis of the Society, has announced the following exposition committees: *Advisory Committee:* H. P. Gant, Chairman, Philadelphia; A. S. Armagnac, New York; D. S. Boyden, Boston; W. H. Carrier, Newark, N. J.; A. C. Edgar, Philadelphia; Roswell Farnham, Buffalo; C. V. Haynes, Philadelphia; E. B. Langenberg, St. Louis; J. I. Lyle, Newark, N. J.; J. F. McIntire, Detroit; H. C. Murphy, Louisville; F. R. Still, New York; E. K. Webster, Camden, N. J.; and H. L. Whitelaw, New York.

Co-operating Committee: O. H. Fogg, president, *American Gas Association;* C. H. Hammond, president *American Institute of Architects;* E. M. Fleischman, president *American Oil Burner Association;* A. J. Wood, president *American Society of Refrigerating Engineers;* Walter Klie, president *Heating and Piping Contractors National Assn.;* H. T. Richardson, president *National Boiler and Radiator Mfrs. Assn.;* John W. Meyer, *National District Heating Assn.;* W. C. Hanson, president *National Pipe and Supplies Assn.;* C. E. Hall, director, *National Warm Air Heating Assn.,* and R. T. Creviston, manager, *Plumbing and Heating Industries Bureau.*

The Exposition is under the direction of the International Exposition Co., New York, and Charles F. Roth is personally in charge of the management of the exposition.

A list of the companies which have reserved space in the exposition follows:

Aerofin Corp.
Air Control Corp. of New York.
Air Reduction Sales Co.
Alexander Bros., Inc.
American District Steam Co.
American Gas Association.
American Gas Products Corp.
American Radiator Corp.
Ames Pump Co.
Armstrong Machine Works.
Automatic Burner Corp.
American Oil Burners Corp.
American Warming & Ventilating Co.
Barnes & Jones.
Beckwith Co.
Bishop & Babcock Sales Co.
B-Line Boiler Co.
Brown Instrument Co.
Bryan Steam Corp.
Bryant Heater & Mfg. Co.
Burnham Boiler Corp.
Carrier Engineering Corp.
Century Engineering Corp.
Celotex Co.
Cochrane Co.
Columbia Burner Co.
Combustion Fuel Oil Burner Co.
Combustion Specialties Corp.
Cooling & Air Conditioning Corp.
Cook Electric Co.
Copper Radiator Sales Corp.
Coppus Engineering Corp.
Crane Co.
Chicago Pump Co.
Cox, Abram Co.
Crystal Oil Burner Corp.
Culbert-Whitby Co.
Direct Control Valve Co.
Domestic Stoker Co.
Dunham Co., C. A.
Durston Gear Corp.
Economy Oil Burner Co.
Economy Pumping Machinery Co.
Engrg. Publications Inc.
Everhot Heater Co.
Flagg & Co., Stanley G.
Frank Heater & Eng. Co., O. E.
Foxboro Co.
Fuel Oil & Temperature Journal
General Electric Co.
General Gas Light Co.
Grinnell Co., Inc.
Hardinge Bros.
Hart & Hutchinson Co.
Hartzell Propeller Co.
Heating Journals, Inc.
Heating and Ventilating.
Heggie-Simplex Boiler Co.
Hill Company, E. Vernon.
Hoffman Specialty Co., Inc.
Hirschman, W. F. Co.
Illinois Engineering Co.
Iona Ventilator Co., Inc.
International Burners Corp.
Janette Mfg. Co.
Johnson Company, S. T.
Johnson Service Co.
Kelly Brass Works.
Kewanee Boiler Corp.
Kieley & Mueller, Inc.
Korfund Co., Inc.
Langenberg Mfg. Co.
Leslie Co.
Linde Air Products Co.
Liquidometer Corp.
Marsh & Co., James P.
McDonnell & Miller.
McIllvaine Burner Corp.
McQuay Radiator Corp.
Mears-Kane-Ofeldt, Inc.
Mercoid Corp.
Minneapolis-Honeywell Reg. Co.
Modine Mfg. Co.
Molby Boiler Co.
Moterstoker Corp.



COMMERCIAL MUSEUM OF PHILADELPHIA

Monarch Metal Weatherstrip Co.
McCord Radiator & Mfg. Co.
Multicell Radiator Co.
Meyer Furnace Co.
May Oil Burner Corp.
Mueller, L. J. Furnace Co.
Nash Engineering Co.
National Regulator Co.
Nesbit, Inc., John J.
Newport Boiler Co.
National Radiator Co.
National Warm Air Heating Ass'n.
O. E. Specialty Mfg. Co.
Orr & Semtower, Inc.
Parks-Cramer Co.
Penn Electric Switch Co.
Pennsylvania Furnace & Iron Co.
Piqua Electric Mfg. Co.
Rawlplug Co., Inc.
Raymond Co., F. I.
Richmond Radiator Co.
Rome Brass Radiator Corp.
Roper Corp., George D.
Ric-Wil Co.
Richardson & Boynton Co.
Sarco Company, The.
Schade Valve Mfg. Co.
Sheffler-Gross Co,. Inc.
Skinner Bros. Mfg. Co. Inc.
Silent Automatic Corp.
Skidmore Corp.
Smith Twin Tubular Boiler Co.
Spencer Heater Co.
Sprayo-Flake Co.
Star Expansion Bolt Co.
Stickle Steam Specialties Co.
Strandwitz & Scott, Inc.
Sweet & Doyle Foundry & Mach. Co.
Sterling Engineering Co.
Sheer Company, H. M.
Schutte & Koerting Co.
Staynew Filter Corp.
Taco Heaters, Inc.
Thatcher Co.
Time-O-Stat Controls Co.
Trerice Co., H. O.
Thermal Units Company.
United States Gypsum Co.
United States Radiator Co.
Vinco Co., Inc.
Warren Webster & Co.
Webster Tallmadge & Co., Inc.
Westinghouse Elec. & Mfg. Co.
Wilcolator Co.
Wilson Bros.
Wood Conversion Co.
Wright Austin Co.
Winchester Repeating Arms Co.
York Heating & Ventilating Co.
Young Radiator Co.
Young Pump Co.
Yale & Towne Mfg. Co.

New Plant For O-E Specialties

L. F. Ostrander, president of O-E Specialty Mfg. Corp., Milwaukee, Wis., announces that increased facilities have been provided for the manufacture of O-E heating specialties and the new plant of the company is now located at 1710 St. Paul Ave.

Eighth National Power Show

Every manufacturer and producer has in mind keeping his production costs at a minimum. This power plant efficiency is possible through the use of various types of equipment designed for that purpose, most important of which are instruments.

Instruments have achieved a place of importance in the operation of power plants which apparently is not fully appreciated.

At the eighth National Exposition of Power and Mechanical Engineering, known as the National Power Show, to be held in Grand Central Palace, New York City, during the week of December 2nd the manufacturer and operating engineer will find a greater variety of latest equipment than could be viewed in an extensive tour through scores of the most interesting industrial plants in the country.

It is a well known fact that many power plants, boiler rooms and mechanical departments are almost hopelessly out of date, that the owner's operating costs are mounting and profits correspondingly decreasing.

To superintendents and operating engineers of such plants the National Power Show will have a special appeal as a means of informing themselves on the latest phases of power plant equipment and by comparison with their own plant needs be in a position to go to their superiors with a definite plan of rehabilitation, thereby establishing savings in operating costs and more satisfactory operation.

The National Power Show serves as a post graduate course for manufacturers, superintendents, mechanical, heating and ventilating engineers and others in charge of production who are desirous of keeping abreast of the times in engineering progress and by discussion, examination and observation becoming conversant with the last word in equipment which is productive of mental and financial benefit.

At the Power Show one meets the men who are responsible for production in all types of plants.

The days of secrecy in industrial plants are relics of an almost forgotten past. There was a time when plant visitors were about as welcome as the proverbial insurance solicitor. But things have changed. There is a new era of open mindedness. The factory man today is glad to have you drop in because he feels that he will get as much out of the interview as you do.

At the Power Show producers, consumers, engineers, superintendents and others in the field meet on a common ground and in establishing a sort of business and social relationship, mutual benefit results.

Illustrating the diversified field of visitors who make an annual pilgrimage to the Power show, 26,065 out of last year's attendance registered. They came from 1056 cities and towns in the United States and 109 towns and cities in 36 foreign countries.

An analytical survey of those who registered showed 6,989 executives; construction men, including architects, builders, engineers, designers, contractors and superintendents, 2,867; highly trained engineers and their assistants, 4,722; operating men, including managers, department heads, engineers, etc., 9,040, while the educators, foreign visitors and government officials numbered 1,118.

These visitors represented 78 groups of industries, indicating 545 types of products manufactured, industries or activities in industrial pursuits.

The 420 odd booths already subscribed for in the Power show will occupy approximately 130,000 square feet of surface on three floors of Grand Central Palace.

The men who will be in charge of the individual exhibits have been chosen for their knowledge of the products and for their ability to explain and assist those desiring information.

The range of equipment being shown this year is the most extensive in the eight years of the exposition's history and will include practically every branch of machinery and equipment in the power generation and mechanical engineering field.

The American Society of Refrigerating Engineers will hold its annual meeting during the progress of the Power Show and the *American Society of Mechanical Engineers* will also be in session in New York during the week of December 2nd.

Charles F. Roth and Fred W. Payne, with offices in Grand Central Palace, New York City, are in charge of the National Power Show management.

The Advisory Committee of the National Power Show is made up of the following well known engineers: I. E. Moultrop, *chairman,* Edison Electric Illuminating Co., Boston; Homer Addams, past president, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; A. Black, *chairman,* Professional Divisions, *A. S. M. E.;* N. A. Carle, *General Manager* Pacific Electric & Manufacturing Co.; Fred Felderman, past National President, *National Association of Stationary Engineers;* F. M. Gibson, *chairman,* Power Division, *A. S. M. E.;* C. F. Hirshfeld, chief of Research Department, Detroit Edison Co.; O. P. Hood, Chief Mechanical Engineer, United States Bureau of Mines; John H. Lawrence, Thomas E. Murray, Inc.; Thornton Lewis, President, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; Fred R. Low, past President the *American Society of Mechanical Engineers;* David Moffatt Myers, Consulting Engineer; M. S. Sloan President, *National Electric Light Association;* Elmer A. Sperry, President, *American Society of Mechanical Engineers* and Arthur J. Wood, President, *American Society of Refrigerating Engineers.*

Local Chapter Reports

Cleveland Chapter

The Cleveland Chapter of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS has had a very active year, showing a status of membership of 36 members.

The speaker at the October meeting was R. H. Conner, director of the Cleveland Research Laboratory of the *American Gas Association,* who gave a talk on the extensive use of gas fuel for heating.

E. E. Kelly discussed insulation as applied to building construction at the November meeting.

The next meeting was the January 1929 meeting, at which Prof. S. E. Dibble of Carnegie Institute of Technology, and Horace Wetzel, Cleveland, addressed the members on pipe welding as applied to the heating business.

The February meeting was held in conjunction with the Cleveland Engineering Society, and J. W. Bishop gave an interesting talk on the Ford museum.

At the March meeting the Chapter members were honored by a visit from Thornton Lewis, president of the society, Prof. A. C. Willard, University of Illinois and past-president of the society, A. V. Hutchinson, secretary, and several members of the National Council. Talks of considerable interest were delivered by President Lewis and Professor Willard, and others on the Council.

A talk on district steam heating was given at the April meeting by Ernest D. Dubry, assistant superintendent of central heating of the Detroit Edison Co.

The next meeting was held on June 14, and was for the purpose of the annual election of officers. Following the regular order of business there was a smoker and a special hour of bridge.

According to Secretary Davis' report, the Cleveland Chapter anticipates a talk by Max Rather of the Johnson Service Co., Milwaukee, Wis., during the coming year, but no other plans have been made at the present time.

The following officers are at present serving the Cleveland Chapter:

President, W. C. Kammerer.
Vice-President, F. H. Morris.
Secretary, R. G. Davis.
Treasurer, H. M. Nobis.
Board of Governors, C. F. Eveleth, F. A. Weager and D. E. Humphrey.

Kansas City Chapter

A very successful year has just been experienced by the Kansas City Chapter of the Society, judging from the yearly report of the Secretary, F. A. Kitchen. He also advises that the success and progress during this past year was due to the untiring efforts of Herman Henrici, president.

The membership of the Society was maintained at its previous level and the influence of its activities was more greatly felt in this territory. The number of members of the Kansas City Chapter in good standing is 38.

One of the first subjects to be discussed, during the year, was the Modernization of Homes, and a correlative talk showing the connection between the Society and the modernization movement. Then followed an address on the subject of Soundproofing of Buildings as applied to heating, ventilating and refrigeration installations, and then electricity as applied to heating, ventilating and air conditioning equipment. Later the subjects of Warm Air Heating and the Ventilation of the Vehicular Tunnel were discussed, followed by oil burning in the heating industry. A descriptive talk on the History of Heating and Ventilation in this country was another interesting topic discussed, as well as the Society's Guide, and the welding of pipe as applied to heating.

September 9, 1929. The regular September meeting of the Kansas City Chapter of the Society was held on Monday evening, September 9, 1929, at the Ambassador Hotel, with an attendance of 31 members and guests.

The subject of the evening, which was *Gas, Its Supply, Distribution and Use,* was presented by Mr. Patten and Mr. Koinn of the Kansas City Gas Co., a subsidiary of the City Service Co. Mr. Patten described in detail the abundant supply of gas for this section of the country, as secured in the large gas sections of Oklahoma and Texas. A graphic description of the piping systems connecting the states of Texas, Oklahoma, Missouri and Kansas was given.

In view of the present increased use of gas as a heating medium in Kansas City, this discussion came at an appropriate time and the discussions arising from this subject were extremely interesting. The questions of gas pressures, adequate piping systems within Kansas City and proper equipment for burning natural gas were brought up in detail.

The following interesting topics have been scheduled for the coming year: Gas, its supply, distribution and use; The future of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS; Heat losses in connection with test data; Power plant equipment for high pressure; Air conditioning; THE GUIDE AND RESEARCH of the Society; Warm air heating practice; and lastly the annual meeting of the Chapter.

The present officials of the Kansas City Chapter are as follows:

President, C. C. Clegg.
Vice-President, E. K. Campbell.
Secretary, F. A. Kitchen.
Treasurer, C. W. Plass.
Board of Governors, Henry Nottberg, Ben Naylor and C. C. Clegg.

The following Committees have been appointed by President Clegg: *Chapter Relations Committee,* E. K. Campbell, *chairman,* W. F. Cox, B. F. Cook, Ernest Jones; *Membership Committee,* Carl Weiss, *chairman,* W. E. Gillham, J. G. Lewis, J. H. Kitchen; *Publicity Committee,* Amdi Worm, *chairman,* F. A. Griffin, David Caleb; *Civic Committee,* Nate W. Downes, *chairman,* J. M. Arthur, H. H. Wright; *Welfare Committee,* John Fehlig, *chairman,* Linn W. Millis, F. P. Hitchcock.

President Clegg anticipates great progress for the ensuing year.

Illinois Chapter

The Illinois Chapter of the Society has had a year of successful meetings, closing with a total membership of 174.

H. L. Fogleman was the speaker at the October meeting, the topic of which was The Engineer; Should He Be a Salesman? At the November meeting, H. B. Johns, manager of House Heating division of Peoples Gas, Light and Coke Co., delivered an address on Gas—Its Manufacture, Storage and Distribution, and Its Use in Space Heating. Trouble Shooting was the general discussion which took place at the December meeting. F. E. Hartman, chief chemist of the U. S. Ozone Co., selected Ionization and Its Relation to Ventilation and Health as his subject for discussion at the January 1929 meeting. Prof. A. C. Willard, University of Illinois, and past-president of the Society, rendered an address on Research—Fundamental and Applied. The method used in installing the new ventilating system in the Houses of Congress at Washington, D. C., was the subject in discussion at the March meeting, which was lead by J. I. Lyle, treasurer

and general manager of Carrier Engineering Corp. The speaker at the April meeting was Prof. L. E. Davis, of the mechanical engineering department of Armour Institute, who spoke on the Method for Measuring Air Quantities delivered by a ventilation system to rooms to be occupied by human beings, the method to include measurements of air quantities at the register faces.

No definite plans have been made for the coming year further than the October meeting which will be addressed by Thornton Lewis, president of the Society.

The members now holding office in the Chapter are as follows:

President, H. G. Thomas.
Vice-President, T. H. Monaghan.
Secretary, C. W. DeLand.
Treasurer, August Kehm.
Board of Governors, J. H. O'Brien, R. B. Hayward and E. P. Heckel.

The nominating committee consisting of W. A. Mertx, *chairman,* L. L. Narowetz, Jr., and G. H. Blanding is the only committee now serving.

In line with their recommendation, the following officers are nominated for the coming year: *President,* H. G. Thomas; *Vice-President,* T. H. Monaghan; *Secretary,* C. W. DeLand; *Treasurer,* C. W. Johnson. *The Board of Governors* nominated consists of J. H. O'Brien, M. S. Good and E. P. Heckel.

Massachusetts Chapter

A very successful year has just been completed by the Massachusetts Chapter of the Society, which is shown in the annual report of the Secretary.

All the meetings of the year were held at the Engineers Club, Boston, and each was preceded by a Get-Together session and dinner.

The first meeting of the Chapter year was the October 1928 meeting, at which A. H. Barker spoke on the engineering improvements of the Boston and Maine R. R., for which he is assistant publicity manager. Motion pictures of the 1927 Vermont flood were also shown.

A. W. Rucker of the university staff, Cambridge, Mass., gave a very interesting address at the November meeting on business analysis as a profit maker.

Prof. A. C. Willard, University of Illinois, and at the time President of the Society, paid a visit to the Chapter and discussed the ventilation problems of the Holland Vehicular Tunnel at the December meeting.

The January meeting had C. E. Mayette of Dwight P. Robinson Co., as speaker. Mr. Mayette discussed the subject of heating, ventilating and refrigeration in the new Boston Garden.

J. C. Grant, engineer for the Plibrico Jointless Fire Brick Co., gave an interesting discussion on Plibrico jointless fire brick at the February meeting.

At the March meeting a lecture was given at the Boston Consolidated Gas Co. Bldg. auditorium by D. S. Reynolds who lectured on the latest developments in domestic gas heating.

The April meeting was put on through the courtesy of Prof. Philip Drinker of the Harvard University School of Public Health. The meeting was divided into two parts—the first speaker being J. M. Dallavalle, research fellow in industrial sanitation at the Harvard Engineering School, who selected as his topic Problems in the Design of Local Exhaust Hoods. The second speaker was Theodore F. Hatch, instructor in sanitation at the Harvard Engineering School, and Harvard School of Public Health. Mr. Hatch discussed the dimensions of the required air velocity for dust control, with particular reference to the granite industry.

The May meeting was the annual ladies meeting held at the Charles River Country Club, Newton, Mass. During the afternoon the ladies played whist, for which suitable prizes were awarded, while the gentlemen played golf. Dinner was served, during which time an orchestra furnished music.

The officers of the Chapter are as follows:

President, T. F. McCoy.
Secretary, J. S. Webb.
Treasurer, L. J. McMurrer.
Board of Governors, T. F. McCoy, G. H. Gleason, Philip Drinker, J. W. Brinton, David Moulton, Lawrence R. Steston and Ernest C. Whittaker.

Minnesota Chapter

The opening meeting of the season of 1928 was held on October 15, at which the members enjoyed an inspection trip through the plant of the Minneapolis-Honeywell Regulator Co. The November meeting was devoted to a general talk by Prof. F. B. Rowley, University of Minnesota. A special meeting was arranged for December and the members were entertained at the Golden Valley Club. Prof. A. C. Willard, professor of mechanical engineering at the University of Illinois and then president of the Society, addressed the members and guests attending the January gathering on the subject of Research in Heating and Ventilating at the University. The reports and a resumê of the papers presented before the Society at the Annual Meeting in Chicago were given at the February meeting of the Chapter. F. C. Houghten, director of the A. S. H. & V. E. Research Laboratory, visited the Chapter at the March meeting and discussed the work of the Laboratory and its influence on the present practice of the heating and ventilating engineers. As well as a talk by Earl Evleth, a moving picture was shown at the April meeting which was entitled Welding of Piping in the Heating, Ventilating and Plumbing Fields. The Chapter concluded its year with their Annual Meeting which took place on May 8, with the reading of reports and the election of officers, all of which followed the annual golf match which aroused great interest among the members.

A tentative program has been outlined for the coming season as follows: It is expected that Thornton Lewis, president of the Society, will honor the Chapter by attending the October, 1929 meeting. An inspection trip through the Flax-li-num Insulating Company's plant is planned for the November meeting. At the December meeting the Boiler Code will be discussed, while the topic for discussion at the January meeting will be on the Oil Burner. Nothing definite has been scheduled for the February meeting, as yet. The March meeting will be devoted to the subject of Copper Radiation. Another inspection trip is planned for the April meeting which will take place in the Mayo Clinic in Rochester. The May, or Annual meeting, will hold the usual interest for the golf enthusiasts of the Chapter.

The officers and Committees serving the Chapter are as follows:

President, E. F. Jones.
Vice-President, D. M. Forfar.
Secretary-Treasurer, M. S. Wunderlich.
Board of Governors, A. J. Huch and W. F. Uhl.

The *Meetings Committee* consists of A. Buenger, *chairman,* A. M. Wagner and R. B. Mosher. Those on the *Membership Committee* are, Leon Hanson, *chairman,* C. E. Gausman and A. Y. Winterer. The *Legislative Committee* is as follows: G. C. Morgan, *chairman,* and O. C. Ruff. The *Nominating Committee* consists of A. J. Huch, *chairman,* A. Buenger and R. W. Otto. H. G. Helstrom and A. L. Sanford are on the *Auditing Committee,* while F. B. Rowley is serving on the *Publications Committee.*

Western New York Chapter

The Western New York Chapter of the Society has completed a very successful year, with a record attendance at every meeting.

Six limited chapter memberships have been recorded during

the past year, and a decided increase in membership is expected during the forthcoming season, according to the Secretary.

The officers, directors and Committees now serving are as follows:

President, O. K. Dyer.
First Vice-President, F. H. Burke.
Second Vice-President, Joseph Davis.
Treasurer, C. H. Love.
Secretary, D. J. Mahoney.
Board of Governors, R. T. Coe, L. A. Harding, C. W. Farrar, Hugo Hutzel, W. G. Fraser, Roswell Farnham, M. S. Jackson and C. A. Evans.
Speakers Committee: Hugo Hutzel, chairman, C. W. Farrar and W. G. Fraser. *Entertainment Committee*: Joseph Davis, *chairman,* R. T. Thornton, J. J. Landers. *Membership Committee*: F. H. Burke, *chairman,* William Roebuck, Jr. and J. J. Yager.

The general election of officers for the new year took place at the January meeting of the Chapter. The new officers were installed at the February meeting which was held at the Hotel Buffalo, where a dinner dance had been arranged by the members. Hugo Hutzel rendered a discussion at the March meeting on the subject of Latest Developments in the Use of Pulverized Fuel, and M. F. Rather, Cleveland, spoke at the April meeting on the Latest Achievements in Automatic Temperature Regulation. The May meeting proved to be most interesting, as the Chapter had as its guests the Senior Engineers from the University of Kentucky, as well as the National officers of the Society. The Western New York Chapter concluded its season with a summer party for the June meeting which was held at the Automobile Club.

Michigan Chapter

Considerable progress has been made during the past year by the Michigan Chapter of the Society, which shows a membership of 109 members and 91 limited chapter members, according to Secretary Clark's record.

At the October meeting Prof. R. S. Hawley discussed the Extension Work at the University of Michigan, and W. R. Rhoton was the speaker of the November meeting. Warm Air Heating was the topic of discussion at the December meeting which was rendered by J. C. Miles. H. L. Walton selected Air Conditioning System in the Union Trust Building in Detroit as his subject for discussion at the January 1929 meeting. The members and guests present at the February meeting enjoyed the speaker of the evening, J. W. Parker, the chief engineer of the Detroit Edison Co. Thornton Lewis, president of the Society, honored the Michigan Chapter members by attending their March meeting, which was thoroughly enjoyed by the members and guests attending. The subject chosen for the April meeting was Heating, Ventilation and Air Conditioning Systems in the Fisher Building in Detroit, which was discussed by Herbert Ziel of Albert Kahn's Inc., G. D. Winans, of the Detroit Edison Co., and Fred Johnson of Johnson-Larsen & Co., heating and ventilating contractors. The May meeting was devoted to the annual election of officers and a golf meet.

A tentative schedule has been planned for the coming year, and will be, if possible, as follows: A visit from L. A. Harding, vice-president of the Society, and Past Presidents' Night is the arrangement for the meeting to be held on October 14th. Coal, Oil and Gas will be the discussion at the November 11th meeting. A Keno party has been scheduled for the December meeting which will be held on the 16th. A discussion on the Relationship which should exist between the Architect, Engineer and Contractor will be given by A. W. Barron, engineer, H. W. Meier, architect, and Fred Johnson, heating and ventilating engineer, at the January 13th meeting. The February meeting will be devoted to Detroit Edison Night. The AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS derby will take place at the May meeting to be held on the 17th. W. H. Carrier, Carrier Engineering Corp., and C. W. Farrar, Excelso Products Corp., Buffalo, will be the speakers at the April 14th session. The concluding meeting, May 12th, will be given over to the annual election of officers and golf meet.

The officers, directors and committees now serving are as follows:

President, W. G. Boales.
Vice-President, E. E. Dubry.
Treasurer, R. K. Milward.
Secretary, E. H. Clark.
Board of Governors, L. L. McConachie, W. J. Whelan, H. M. Stephen.
Program Committee, W. J. Whelan, *chairman;* N. B. Hubbard, *vice-chairman; Membership Committee,* G. H. Giguerre, *chairman;* H. M. Stephen, *vice-chairman.*

Pacific Northwest Chapter

Many profitable meetings were held during the past season by the Pacific Northwest Chapter, at which subjects of great value were discussed.

Among the various topics selected for discussion during these meetings were the following: Orifice system of heating; ventilation in tunnel construction; state heating and ventilating code; status of the heating and ventilating engineer.

The officers and committees now serving the Chapter are as follows:

President, E. L. Weber.
Vice-President, E. O. Eastwood.
Secretary, M. Anderson.
Treasurer, W. W. Cox.
Board of Governors, W. L. Dudley, C. F. Twist and W. E. Beggs.
Membership Committee, F. B. Delong, P. M. O'Connell and W. E. Beggs; *Program Committee,* W. E. Beggs and W. W. Cox.

Thirty-three members enrolled in the Pacific Northwest Chapter of the Society up to September 1, 1929.

Philadelphia Chapter

The Philadelphia Chapter has shown great progress during the year just passed, which is obvious from the report of the Secretary, L. C. Davidson. The total membership on September 1, 1929 was 177, comprising 136 members, 34 associate members and 7 junior members. In addition there are 6 limited Chapter members.

The January meeting, which was the annual Chapter meeting, was devoted entirely to the election of officers for the year 1929, after which the meeting was given over to entertainment and a general get-together.

L. G. Paine addressed the members at the February meeting on the subject of Application of Sub-Atmospheric Pressure to Steam Heating.

Lucian Buck was the speaker at the March meeting, selecting as his subject the Industrial Drying Problems.

At the April meeting J. I. Lyle, Carrier Engineering Corp., gave an interesting talk on Air Conditioning System Installed in the U. S. Capitol at Washington. This meeting was very well attended, and was, perhaps, the most popular meeting of the year.

The members and guests assembled at the Plymouth Country Club for the May meeting, where they enjoyed golf, tennis and other outdoor sports during the afternoon, and in the evening they visited the Frost Research Laboratory.

During the months of June, July, August and September no meetings were held.

A tentative schedule of papers to be discussed during the coming Chapter season is as follows: October, a moving picture en-

titled the Wonders of Anthracite will be shown; for the November meeting the subject of Heating with Gas will be discussed; Ladies' Night is scheduled for the December meeting. The annual election of officers will take place at the January meeting, together with a special business session. Heating with electricity has been chosen for the topic to be discussed at the February meeting.

The speakers for the other meetings of the year have not, as yet, been scheduled, while the schedule just given is tentative and will be subject to change.

The Chapter officers are as follows:

President, A. C. Edgar.

Vice-President, H. G. Black.

Secretary, L. C. Davidson.

Treasurer, E. N. Sanbern.

The above Chapter officers, including the following three men, constitute the *Board of Governors,* R. V. Frost, A. J. Nesbitt, Lee Nusbaum.

The various committees with the chairmen are as follows: *Meetings Committee,* Merrill F. Blankin; *Publicity Committee,* R. C. Bolsinger; *Membership Committee,* W. A. Bornemann; *Committee on Attendance,* W. G. Culbert; *Finance Committee,* Lee Nusbaum; *Year-Book Committee,* C. D. Graham.

Pittsburgh Chapter

A series of interesting meetings have been enjoyed by the members of the Pittsburgh Chapter of the Society during the past season, and according to Secretary Teague the Chapter members were very fortunate in having such a great variety of subjects of exceptional value.

October 1, 1928—Open House at the Research Laboratory, followed by a meeting at the William Penn Hotel where the following prominent members made brief remarks concerning affairs of the Society and of the Research Laboratory: A. C. Willard, S. R. Lewis, Thornton Lewis, L. A. Harding, W. H. Carrier, F. B. Rowley, J. F. Hale, H. P. Gant, F. D. Mensing, H. H. Angus, W. A. Rowe and A. P. Kratz.

November 5, 1928—Prof. A. C. Willard, then president of the Society, spoke on his research activities in connection with the ventilation of the Holland Tunnel.

December 3, 1928—W. E. Stark of the Bryant Heater & Mfg. Co., Cleveland, talked on the subject of heating with gas with special reference to heating with hot water and steam.

January 7, 1929—Percy Nicholls, head of the fuel section at the U. S. Bureau of Mines and a member of the boiler code committee of the Society, discussed the code for the Rating of Low Pressure Boilers and the problems involved.

February 11, 1929—Dr. R. E. Hall, director of Hall Laboratories, Inc., specialists on boiler problems, and formerly physical chemist of U. S. Bureau of Mines in charge of investigations on boiler problems, discussed some characteristics of corrosion and of the boiling of water. He discussed the effect of O_2 and CO_2 on corrosion and effect of calcium and other salts on the boiling of water.

March 11, 1929—J. M. Robb of the Herman Nelson Corp., spoke on Unit Ventilation, giving very complete information on the history, development and present day usage of this phase of heating and ventilating.

April 8, 1929—Thornton Lewis, president of the Society and president of the York Heating and Ventilating Corp., Philadelphia, gave a very interesting address to the members and guests on the operation of the Society and plans for the coming year.

May 6, 1929—A motion picture film showing the newest developments in pipe welding was procured. Prof. S. E. Dibble of the Carnegie Institute of Technology under whose supervision the film was prepared talked on this subject. In addition A. L. Weixel spoke on research at the Armour Institute on the accurate use of anemometers.

The officers and committees now serving the Pittsburgh Chapter of the Society for the year 1929 are as follows:

President, H. Lee Moore.

Vice-President, E. W. Stitt.

Secretary, W. W. Teague.

Treasurer, Peter O'Neill.

Board of Governors, F. A. Gunther, R. H. Heilman, M. M. Timmerman.

Attendance Committee—F. A. Gunther, chairman; H. E. Digby, E. C. Evans, G. M. Comstock, H. A. Beighel, E. W. Stitt. *Program Committee*—F. C. Houghten, *chairman;* Roy Brauer, M. M. Timmerman. *Membership Committee*—C. W. Wheeler, *chairman;* F. C. McIntosh, Thomas M. Dugan.

The status of membership on September 1, 1929, was as follows: 48 members, 11 associate members and 3 junior members, making a total of 62 members in the Pittsburgh Chapter.

Wisconsin Chapter

The Wisconsin Chapter of the Society has concluded a very successful year with 49 comprising the Chapter membership.

At the October meeting in 1928 the officers for the ensuing year were elected, and plans for the coming year were discussed.

E. A. Jones, one of the Chapter members, gave an interesting talk on Gas Heating which was enjoyed by those members and guests who attended the November meeting.

A bowling party was arranged for the December meeting, together with a discussion on methods of securing new members for the Society.

At the January 21st meeting W. M. Richtmann addressed the members on the Effect of Frame Calking and Storm Windows on Infiltration Around and Through Windows, which was discussed and enjoyed by those in attendance.

C. E. Bronson, Kewanee, was the speaker at the March meeting, choosing for his topic, Boilers, Smoke Breeching and Chimneys. This meeting was dedicated to the past presidents of the Chapter, four of whom were present.

The subject of discussion at the April meeting was Pipe Welding in the Heating and Ventilating Field, which was rendered by Mr. Evleth and proved very interesting.

The May meeting was held at the plant of the James Mfg. Co., Fort Atkinson, Wis., where the members and guests were given the opportunity to inspect the plant. Following the inspection Mr. Shodron and Mr. James, both of the company, addressed the members and guests attending.

A picnic for the members and their guests was arranged for the June gathering.

The following members are now holding office in the Wisconsin Chapter:

President, Wm. F. Noll.

Vice-President, Fred G. Weimer.

Secretary, V. A. Berghoefer.

Treasurer, Martin Erickson.

The present *Board of Governors* is as follows: H. M. Miller, Paul C. Downey and G. L. Larson.

CANDIDATES FOR MEMBERSHIP

The Constitution of the Society, as now amended, requires the following mode of procedure in voting on applicants for membership in the Society. All applications for membership are to be sent to the Secretary and the name of applicants and their references shall be printed in the next issue of the Journal of the Society or sent to the members in other approved manner as ordered by the Council. When replies are received from references, the Candidate's application shall be submitted to and acted upon by the Membership Committee as soon as possible.

When the Membership Committee has acted favorably upon a Candidate's application and assigned his grade the Council shall vote upon the election of the proposed Candidate for membership by letter ballot. During the past month 4 applications for membership have been received and the names of these men and their sponsors are published in the following list.

Members are requested to scrutinize the list with care. The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into membership by advising the Secretary promptly of any whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote.

Unless objection is made by some member by November 1, 1929, these candidates will be balloted upon by the Council. Those elected to membership will be notified by the Secretary, immediately after election.

CANDIDATES	REFERENCES	
	Proposers	*Seconders*
RAYMOND, FRED I., President, F. I. Raymond Co., Chicago, Ill.	Wm. H. Chenoweth H. G. Thomas	Macy S. Good C. A. Thinn
SIMPSON, RICHARD BALFOUR, Asst. Mgr., Building Dept., The Celotex Co., Chicago, Ill.	P. D. Close J. H. Bracken	Wm. H. Moler C. W. Presdee
SIMMS, ROBERT C., Dist. Supt. of Construction, Carrier Engrg. Corp., Philadelphia, Pa.	M. S. Smith L. L. Lewis	P. L. Davidson W. A. Borneman
WAITE, HARRY, Secy. & Mgr., Gray Plumbing Co., Superior, Wis.	M. H. Bjerken A. J. Huch	Edwin F. Jones Donald M. Forfar

Candidates Elected October, 1929

In the past issues of the Journal of the Society there were posted among the list of Candidates for Membership 16 candidates for election in the different grades of Membership, whose names have been balloted upon by the Council. We are now instructed by the Council to post herewith, as required by Art. II, Sec. VIII, of the By-Laws, the following list of candidates elected:

MEMBERS

CORNELIUS, FRANK HUDSON, Cornelius Engrg. Co., 7400 Idlewild St., Pittsburgh, Pa.

DAVIS, OTIS E., Sales Engr., Hoffman Specialty Co., Scottsbluff, Nebr. (*Advancement.*)

FORTUNE, J. ROBERT, Wickes Boiler Co., Saginaw, Mich.

GROSS, SAMUEL, Treas., Sheffler-Gross Co., Inc., 203-11 Drexel Bldg., Philadelphia, Pa.

GURNEY, EDWARD HOLT, Pres., The Gurney Foundry Co., Ltd., 500 West King St., Toronto, Ont., Canada.

KILNER, JOHN SAUNDERS, Partner, Kilner-Mills Co., 3-266 General Motors Bldg., Detroit, Mich.

JARDINE, DOUGLAS CONNELL, Jardine & Knight Plumbing & Heating Co., 312 Custer Ave., Colorado Springs, Colo. (*Advancement.*)

MALONE, DAYLE G., Vice-Pres. & Genl. Mgr., Hardin-Lavin Co., 121 W. 39th St., Chicago, Ill. (*Advancement.*)

ASSOCIATES

BERNHARD, GEORGE, Heating Engr., George Bernhard Heating Co., Inc., 1007 Church Ave., Brooklyn, N. Y.

FREEMAN, ALTON M., Sales Engr., H. M. Miller, Milwaukee, Wis.

HARDINGE, FRANKLIN, Pres. & Genl. Mgr., Hardinge Bros., Inc., 4149 Ravenswood Ave., Chicago, Ill.

LAWSON, WILLIAM IRVIN, Vice-Pres. & Genl. Mgr., The Buckeye Blower Co., Columbus, Ohio.

PORZEL, JOSEPH, Vice-Pres., Dir. Research, Multicell Radiator Co., Buffalo, N. Y.

SHAVER, HERBERT H., Asst. Genl. Sales Agent, Hudson Coal Co., Scranton, Pa.

STITT, HOWARD B., Sales Engr., National Radiator Corp., 431 W. Georgia St., Indianapolis, Ind.

STROCK, CLIFFORD, Asst. Editor, Heating and Ventilating, 521 Fifth Ave., New York, N. Y.

JUNIORS

GUNTHER, RAYMOND CHESTER, Sales Engr., Carrier Engrg. Corp., Land Title Bldg., Philadelphia, Pa.

HAKES, LEON MARC, Sales Engr., The R. T. Coe Cos., 907 Gas & Electric Bldg., Rochester, N. Y.

KUEMPEL, LEON L., Sales Engr., Minneapolis-Honeywell Regulator Co., 2920 4th Ave., S., Minneapolis, Minn.

SCHWARTZ, JACOB, 30 West 27th St., Bayonne, N. J.

Heating · Piping
and Air Conditioning

EDITORIALS

Pipe Line Expansion

Engineering is an exact science. Yet it deals with many phenomena that are pretty difficult to interpret in such an exact fashion that it is possible to say—this is it.

Take pipe line expansion as an example that has come to our attention editorially in this issue. There are many tables and charts purporting to give information on the amount of expansion that will take place in pipe lines at given temperature differences—each one trying to deal with this phenomenon in an exact way. Yet there is lack of agreement among them, and the status of pipe line expansion, therefore, is apparently anything but exact.

We published a simple method of determining pipe line expansion over a range of temperature differences from 0 to 800 deg. fahr. A reader who is recognized as an expert in the piping field submits another method that proves fallacies in the first, but, at the same time, does not insist that the method he uses is absolutely exact. At least he suggests that we endeavor to get a consensus of opinion from those who have occasion to use such tables in the hope that out of it something authoritative on the problem might evolve.

This reader's discussion is presented in this issue. If there be confusion and uncertainty as a result of various methods aiming to give the same information on pipe line expansion, this is a step toward clearing it all up.

International Events

In science, the peoples of the world have a universal language; in the practical application of science—engineering—the inhabitants of the earth have a common understanding. No one, no matter of what nationality, could help admiring and thrilling at the engineering skill that made Lindbergh's historic flight possible. French, Japanese, Russians and Americans are just as proud of the designers and navigators of the Graf as are the Germans. Accomplishments of this kind are accomplishments that we, as citizens of the world and not as citizen of any one country, may applaud and build on in the future. Engineering is international in character; it transcends political boundaries.

It is fitting, therefore, that engineers of every nationality should meet together in the interests of the profession, as they are doing this month in Tokio at the World Engineering Congress. Mutual respect and understanding between the engineers of every nation in attendance will inevitably follow. So, too, will the first International Heating and Ventilating Exposition in Philadelphia next January result in strengthening the already numerous bonds between the engineers of the world. And, with our industrial development, engineers are becoming the leaders in every field today. Such events will have a far-reaching influence.

Aids to Design

Charts, tables and easily worked formulae that take the hours and mistakes out of figuring are to the engineer what labor saving machinery is to the producer. Some enterprising statistician could probably demonstrate that millions of hours are saved annually in the engineering profession by a curve on a chart or by a column of figures in a table. The same statistician might be even more convincing in his proof of the dollars saved in the mistakes that are avoided by these aids to figuring. So it is little wonder that the engineer and contractor compile their little "bibles" of technical press clippings, catalog data and what not so as to have readily available the answer to a problem of design or installation.

Pages of HEATING, PIPING AND AIR CONDITIONING are taking their places in these "bibles." We recommend for an especially prized position the charts in the article "Sizing Chimneys for Tall Buildings" appearing in this issue. The figuring and investigating that went into the preparation of these charts by Mr. Hattis were painstaking and thorough. The charts that result are said to give in minutes an answer that would otherwise take hours of figuring.

Then, for the heating and piping contractor and engineer, the charts for determining proper pipe sizes in "Economical Pipe Combinations in Heating Systems" will be a welcome addition to the "bible." Also, the discussion on pipe line expansion includes a chart that will help the designer check his figures quickly when he must determine the allowance to make for the phenomenon of expansion as temperatures go higher and higher. And in several other articles formulae are given that will guide figures along the right road to a desired solution.

Certainly the more of these charts, tables and formulae that are made part of the working tools of the designer and installer the less under-sized and over-

sized jobs there will be to put a black mark against the engineering profession.

Comfort and Health

Human comfort and health—contributing to them, making them better—that is the phase of air conditioning that seems to interest most a great many of the engineers who have made a study of this science. Knowing as they do how much air conditioning can contribute to such comfort and health, they naturally want the public to appreciate its importance to the extent that it will demand it for the home, the office, the school, the amusement center, the stores and any other place where people live or congregate. And there is little doubt that future years will see its universal application, for today there are many examples of its desirability in every such place.

In the past, air conditioning has achieved its prominent position largely because its application to manufacturing processes has carried with it some convincing dollars and cents arguments on why it should be used. Arguments of that sort are not so convincing when the result at stake is human comfort and health. The majority of people are concerned about their health only when they are about to lose it. And all of us put up with a good deal of discomfort during hot, humid summers and take it for granted that we must do it. Therefore, air conditioning for human comfort and health meets greater sales resistance than does air conditioning for manufacturing processes. But it is only a question of time when people will refuse to endure the discomforts and dangers to health that air conditioning can eliminate.

The medical world could be an effective ally. Doctors have not kept pace, generally speaking, with engineers in the development of air conditioning. With engineers able to produce any desired conditions, doctors as a whole do not tell us what those conditions should be, nor do they corroborate as they should the research promoted by the engineers to determine what they should be.

Dry Ice

IN these days of rapid development in every industry, it is essential to keep abreast of the times, and to this end it were well for us to give due consideration to a recent commercial development in refrigeration, solid carbon dioxide. Engineers will watch with interest extended applications of this product.

For more than 50 years one of the laboratory curiosities developed in physics and chemistry has been what is known as solid carbon dioxide.

The general method for developing solid carbon dioxide in the laboratory consisted of releasing liquid carbon dioxide under high pressure into a canvas bag. A percentage of the liquid carbon dioxide thus liberated turned into a very cold snow, while the remainder of the carbon dioxide was converted into a gas, and passed through a canvas bag, thus becoming liberated in the air.

This cold dry snow seemed to hold many advantages, but the price of production under any such processes as were formerly known seemed to prohibit its ever becoming a product of commercial value.

One serious obstacle which formerly prevented so much use of solid carbon dioxide was that of proper insulation in the refrigerator to be used, and also in the shipping container in which the solid carbon dioxide is shipped from the factory to the user. This obstacle has been overcome through improved methods of insulation and greater knowledge of insulated construction.

It is now no uncommon thing to store or ship solid carbon dioxide with as small an evaporating loss as 5 per cent in 24 hours, with practically 200 deg. Fahr. temperature differential between the evaporating point of solid carbon dioxide and the outside temperature of the air.

Developments of the past five years indicate that it is going to play an important role in various applications of refrigeration, and at the present time this seems to be particularly true of long distance transportation of food supplies.

Suffice to say that even now many tons of this snow are made and used every day and such perishable commodities as ice cream are being economically transported from New York and Philadelphia to the more tropical climates such as Cuba in sufficient quantities to create the threat of an embargo on its import.

Carloads of frozen fish may be shipped by rail for five days without re-icing and without thawing. It is thus evident that the future use of solid carbon dioxide as a serious factor in the industry is becoming pretty thoroughly established.

The saving effected by the use of such a convenient refrigerant is evident from the fact that single charges of solid carbon dioxide have maintained perfect refrigeration in a car over a journey normally requiring many re-icings with water ice.

Another use for solid carbon dioxide is in the ice cream delivery truck. It was customary to carry 1,000 lb. of ice and about 150 lb. of salt to refrigerate a truck capable of carrying 300 gallons of ice cream, which is about a 2½ or 3-ton capacity truck. Through the use of solid carbon dioxide a light Ford one-ton chassis may be employed to do the same amount of work.

Just how extensive this revolution in the industry may be and what fields will ultimately be touched and what economic structures affected or overthrown and its ultimate applications is merely, of course, a matter of conjecture at this time.

LEWIS LIPMAN.

"Open for Discussion"

A department in which we follow the custom of technical societies of allotting space in their programs for discussion of the papers presented

HIGH PRESSURE INSULATION

"The Insulation of High Pressure Steam Lines," *by R. H. Heilman.* Page 186, issue of July, 1929

I AM interested in the article on the subject of insulation in the July issue of Heating-Piping and Air Conditioning, written by R. H. Heilman. On page 187, I notice, particularly, the statement "So far, no high pressure insulation has been produced that can be placed on a high temperature line with as high an insulating efficiency as, say, 85 per cent magnesia would give if it would withstand the higher temperatures, and at a cost comparable to the cost of 85 per cent magnesia." Mr. Heilman is probably referring to sectional pipe covering, but the statement is pretty broad. Conduit filler has stood high temperatures of 550 deg. fahr., both in actual practice and in the laboratory, without the slightest deteriorating effect and at these high temperatures the conductivity is about .425.

The cost of conduit fillers installed compares very favorably to the cost of 85 per cent magnesia applied to the pipes, but of course this is only on underground work.

Waterproof conduit fillers are a development that has taken place in the past three years. I believe it would be well worth while for Mr. Heilman to investigate them, and we would be very glad to send any additional information, samples, etc., for testing.

C. GOTTWALD, President,
The Ric-Wil Company

Mr. Heilman Replies

The statement in my article referred particularly to sectional pipe covering as I did not mention in any way any type of underground insulation in this paper, although, as Mr. Gottwald suggests, the inference might be taken that underground insulation could be referred to. However, I feel since this article referred entirely to insulations as applied on steam lines and not used as a filler in conduit that there will be no misunderstanding in regard to the statement.

I am very well aware that there are numerous insulating packings and powders on the market which are used to very good advantage in insulating underground steam lines. I have seen references and have data on many such materials which are sometimes used for this type of construction. It is probable that a very good article could be prepared on the use of insulations for underground systems. I have already written one article on this subject presented before the National District Heating Association.

In regard to the conductivity of conduit fillers, my experience has been that these fillers are very efficient at the lower temperatures but at the higher temperatures met with in present high pressure steam work, such as 800-850 deg. fahr., I have found that practically all packings have a very steep conductivity curve and are, therefore, not as efficient at these higher temperatures as 85 per cent magnesia in combination with high pressure covering, for instance.

R. H. HEILMAN, Senior Fellow,
Mellon Institute of Industrial Research

Another Letter from Mr. Gottwald

I certainly would like to see Mr. Heilman prepare an article on the use of insulation for underground systems.

I remember very well the article he presented before the National District Heating Association at West Baden, Indiana, in 1927 and, to refresh my memory, I have read over this article again and note he stresses the point that the filler type of insulation is being used considerably and no doubt is very efficient where it can be kept absolutely dry. And he says, further, that to keep it dry necessitates care and expense in constructing the conduit in an endeavor to make it absolutely waterproof. At the time he wrote this article the waterproof filler had only been on the market a few months. Since that time waterproof filler has become practically a standard proposition, where filler type of insulation is being used.

I am just mentioning these things to explain what has been happening since 1927, and it makes me believe that the proposed article would be interesting and valuable and bring things up-to-date.

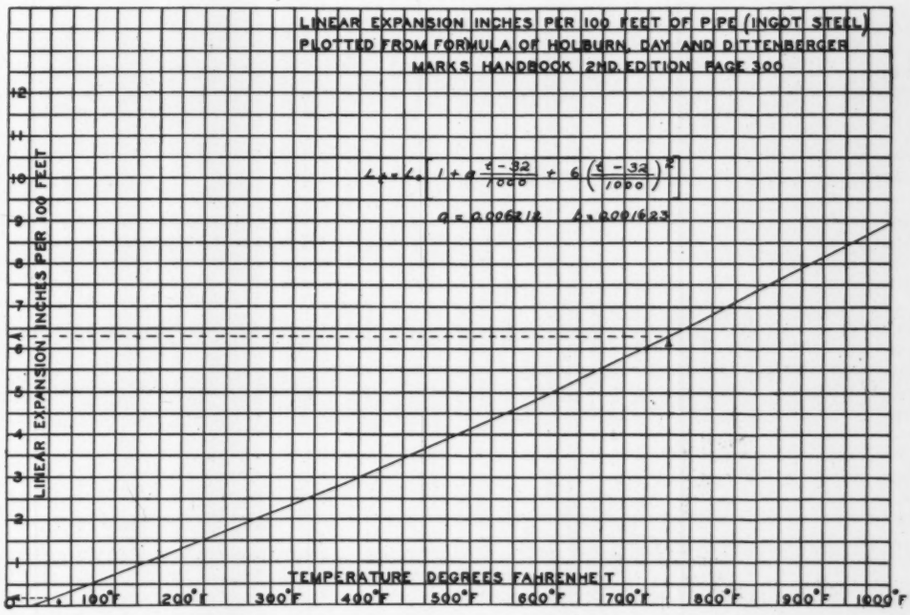
I note that Mr. Heilman's experience with various fillers is that they are very efficient at the lower temperatures and not so good around 800 deg. fahr. Underground systems are seldom, if ever, operated at these higher temperatures. I realize that for power purposes these higher temperatures are used and the insulation must be designed to take care of same. The highest temperature we have tested is 550 deg. fahr., which is the limit of our plate machine. I have, in the past, spent considerable time investigating test reports on conductivity of fillers at high temperatures and after analyzing some of these reports I discovered they were practically worthless and mostly guess work. Some of these fillers were tested at densities that never could be used in practice. Some lack structural strength and can be packed to almost any density, which is not true of the waterproof fillers to which I am referring.

C. GOTTWALD, President,
The Ric-Wil Company

PIPE LINE EXPANSION

A Simple Method of Determining Pipe Line Expansion. Page 301, issue of August, 1929

IN YOUR August issue there appears an unsigned article entitled "A Simple Method of Determining Pipe Line Expansion." Table I of that article gives "Inches of Linear Expansion per 100 ft. of Pipe," for wrought iron, steel and cast iron pipe over a range of "temperature differences" from 0 to 800 deg. fahr. This table is not in agreement with other published data on the expansion of piping and, as interpreted in the example, may prove misleading to one not versed in computing pipe expansion.


LINEAR EXPANSION INCHES PER 100 FEET OF PIPE (INGOT STEEL)
PLOTTED FROM FORMULA OF HOLBURN, DAY AND DITTENBERGER
MARKS HANDBOOK 2ND. EDITION PAGE 300
$L_t = L_o\left[1 + a\frac{t-32}{1000} + b\left(\frac{t-32}{1000}\right)^2\right]$
a = 0.005212 b = 0.001623
LINEAR EXPANSION, INCHES PER 100 FEET
TEMPERATURE DEGREES FAHRENHEIT
100°F 200°F 300°F 400°F 500°F 600°F 700°F 800°F 900°F 1000°F

A Curve for Determining the Linear Expansion of Ingot Steel Pipe

The increasing use of high temperatures in piping work, with the attendant problems of providing flexibility and limiting thrusts on equipment, requires that such a table represent accurately the elongation which will take place for a given change in temperature.

No authority is stated for the coefficients of expansion used in computing the values given in the table. Very evidently, the coefficients used are not in agreement with the generally accepted results of Holborn and Day and others, published by the U. S. Bureau of Standards and in the Smithsonian Physical Tables. The following comparison of published data, abstracted from an article on "How to Lay Out Power-Plant Piping—Section V," by Sabin Crocker, published in "Power," Vol. 64, No. 5, August 3, 1926, indicates the need for careful analysis of such data.

Table I below gives the elongation per hundred feet of pipe for a change in temperature from 77 deg. fahr. to 698 deg. fahr. as calculated from coefficients of expansion obtained from the sources indicated, in comparison with the elongation measured in an actual pipe line.

"The test referred to was made on a section of 16-in. seamless tubing 104 ft. long in the Trenton Channel power house. The line, a steam lead to a main turbine, had been operating for several hours before the final measurements were taken in the hot position, and no further increase in temperature or expansion movement had been observed for some time. Temperatures were read from a thermocouple peened into the outer surface of the pipe wall.

"In view of the care taken in bringing the line up to temperature, and the method used in making temperature readings, it does not seem probable that the observed low coefficient of expansion was due to failure in obtaining the correct temperature on which expansion was figured. It would appear, rather, to check the coefficient determined by Holborn and Day as given in "Mark's Handbook" as coming the closest, among the references quoted, to meeting expansion conditions in an actual pipe line."

Table 1—Linear Expansion Based on Various Coefficients

Material	Linear Exp. In. per 100 Ft. 77 Deg. F. to 698 Deg. F.	Reference	Authority
Steel pipe.....	6.89	Crane Cat. No. 51, p. 593...	None stated
Steel pipe.....	6.88	Marks Handbook.....	None stated
Ingot steel.....	5.49	Marks Handbook.....	Holborn and Day
Steel.....	5.31	Smithsonian Physical Tables, 7th Revised Ed. Reprint, p. 218.....	Holborn and Day
Annealed steel.	4.01	Smithsonian Physical Tables, 7th Revised Ed. Reprint, p. 218.....	Holborn and Day
Steel pipe.....	5.56	Special Test Trenton Channel Plant.....	The Detroit Edison Co. Research Department.

The confusion and uncertainty arising from a multiplicity of tables purporting to give the same information lead one to hope that, if possible, a truly representative table may be found or developed which will cover the temperature range normally encountered and be based on unquestionable authority. Any test results or experience which might be elicited in an effort to substantiate such data should be of great assistance to piping designers.

As an example of the present unhappy state of affairs, consider a 100 ft. length of steel pipe and a change in temperature from 50 to 800 deg. fahr. By the method given in your August issue, the temperature difference is 750 deg. The change in length corresponding to 750 deg. change in temperature is 8.36 inches. The value from the accompanying curve is found as follows: The expansion corresponding to a final temperature of 800 deg. is 6.85 in. The expansion corresponding to an initial temperature of 50 deg. is 0.112 inches, or the expansion of the pipe from 50 to 800 deg. gives an increase in length for 100 ft. of pipe of 6.85—0.112=6.738 inches, approximately 20 per cent less than given in your table.

The method used in your example based on "temperature difference" is sufficiently close for piping having a temperature change from room temperature to some definite operating temperature, but is apt to be misleading as illustrated in the following example: Due to changes in the plant, superheated steam at 700 deg. is to be used in a line originally laid out for saturated steam at 400 deg., and the additional expansion to be cared for must be determined. From your table the expansion to be cared for due to this difference in temperature of 300 deg. would be 2.47 inches, or just the same as a change in temperature from 53 to 353 deg. fahr. It is common knowledge, however, that the coefficients of linear expansion change with increase in temperature, so we know a priori that the result is incorrect.

The method used with the accompanying curve precludes the possibility of this error and is equally simple to apply.

If a consensus of opinion could be obtained from those who have occasion to use such tables, I am certain they would welcome a truly authoritative table, or series of curves giving the linear expansion of piping.

ARTHUR MCCUTCHAN

THE 333 NORTH MICHIGAN BUILDING AT CHICAGO USES A VACUUM HEATING SYSTEM

333 North Michigan Building, Chicago

The mechanical equipment of 333 building, Chicago (60x200 feet ground area, 426 feet above Michigan Avenue, and 43 feet below it), consists of two 205 hp. horizontal return tubular boilers built for 150 lb. working pressure, each equipped with automatic stokers with an overload capacity of 100 per cent.

The ground floor of this building is on the upper level, a double deck street. At the lower level, 43 feet below Michigan Avenue, the coal is received. In the boiler room are two steam driven boiler feed pumps, two steam driven vacuum pumps, one steam driven house pump, one motor driven centrifugal house pump and one motor triplex house pump. There is also one 500 g.p.m. fire pump and equipment. A sprinkler system is provided for the basement.

There are two systems of house tanks; two tanks, one of 2,700 gallons capacity and the other 2,500 gallons capacity located on the twenty-fifth floor serve all floors below the twenty-fourth, and two 1,000-gallon tanks located on the thirty-fifth floor serve all floors above the twenty-third floor.

Two vertical closed hot water heaters serve all floors up to and including the twenty-third floor and the heater on the twenty-sixth floor serves all the floors above the twenty-third. A vacuum system of heating is used. It is down-feed from the twenty-fourth ceiling to the second floor inclusive. The first floor and floors below are served from the main in the basement ceiling. The floors above the twenty-fourth floor are on an up-feed system served from a main in the pipe space above the twenty-fourth floor ceiling.

An exhaust ventilation system has been provided for all interior toilets and all interior office spaces and for certain stores and lower floors.

PRACTICAL PIPING PROBLEMS

Valves—for the Different Classes of Piping and for the Particular Operating and Working Conditions

by W. H. WILSON

Types of Check Valves

SUPPLEMENTING a previous article on valves, appearing in the September issue of this publication, attention is called to check valves, and valves of a similar type.

Standard check valves are made in three regular patterns, depending on the location of the piping on which they are to be installed:—horizontal, vertical and angle. These are of three regular types, lift, swing and ball.

Lift, Swing and Ball

Lift check valves have a disc, with a guiding stem. Their construction is similar to that of globe and angle valves. Some manufacturers use the same valve body for their check valves, of the lift type, as for the corresponding types of globe and angle valves. Some makes have the upper guiding spindle made with a piston head guide that assists in properly seating the disc.

Check valves, when used on compressed air pipe lines, especially on the air discharge line near the air compressor, might pound and chatter. For this purpose, manufacturers supply a check valve of the lift type with a dash pot that causes the check to close slowly, with a cushioning effect.

Lift checks, on account of their construction, restrict the flow somewhat through the pipe line but have the advantage of being easily repaired and reseated. For this reason, in high pressure hydraulic lines and where air and gas lines require valves to hold without leakage, the lift type of check valve is ordinarily used.

Swing check valves, so called from the nature of their construction, having a swinging or hinged gate, offer less restriction to the flow through the pipe lines and are used extensively on water lines. When used on fire protection water lines the insurance people specify the Underwriters' pattern, which has certain requirements, principally a clear and less restricted water way. The larger sizes, which are in the iron body pattern, have brass or bronze trimmings.

Ball check valves are used in the smaller sizes, frequently on oil lines and lubricating connections. Ball check valves, as a rule, are not used on air and gas lines where absolute freedom from slight leakage is required.

Foot Valves

Foot valves are a type of check valve used mostly on pump suction lines and often have a screen or strainer on the inlet, although the Underwriters' type of foot valve, used on fire pump suction pipes, has no screen or strainer. Large check valves and foot valves are made with two or more gates or discs.

Check valves and foot valves for use on cold water lines, or liquids at ordinary temperatures, frequently have the gates and discs faced with leather. For very low pressures a swing check valve, of the horizontal pattern, with the gate hanging nearly vertical, will give good results. This type of a check valve can be used to advantage, in many cases, on low pressure steam heating where exhaust steam is used intermittently.

For gas lines, transmitting low pressure gas, where the pressure is expressed in ounces or inches head of water, and where a check valve or back pressure valve is required for safety and prevention of trouble from back fires, there is a check valve of a special pattern with a very light and sensitive disc, approved by the insurance people having jurisdiction.

Back-pressure Valves

Back-pressure valves are a type of check valve largely used as an atmospheric relief valve for steam engine exhaust pipes, when the exhaust steam is used for heating the boiler feed water, for heating systems or for process work. These valves are either weight- or spring-loaded, and can be adjusted to maintain a certain pressure on the steam supply lines where the heating steam is required. In case the pressure rises above the requirements, the back-pressure valve will open and permit the excess steam to escape to the atmosphere, or to other steam mains and equipment arranged for that purpose, thus preventing the engine from working against an excessive back pressure.

A back-pressure or atmospheric relief valve, used in connection with condensers where a vacuum is maintained, should be arranged to have the valve disc covered with or sealed with water to prevent air from being drawn in if there is a slight leakage, for this would lower the vacuum. In case the vacuum is lost the relief valve will automatically open and relieve the condensing equipment from the excessive pressure. When back-pressure valves are installed in locations requiring the outlet to be piped some distance vertically from the valve, as through the roof of the building, care should be taken to see that a positive drain is provided near the valve so that a slight leakage will not permit water to accumulate

above the valve disc. A head of water may cause sufficient weight to hold the valve shut, requiring an excessive pressure to open it when it is essential that it be opened quickly for safety. Large back-pressure and atmospheric relief valves, such as are used in central power stations in connection with steam power equipment, are frequently of the "down discharge" pattern.

Non-return Valves

Boiler stop and check valves, sometimes called non-return valves, are a type of valve that are widely used on the outlet pipe of a steam boiler. Usually one of the main boiler header valves is a valve of this type. The valve is a type of check valve that will close automatically when a boiler tube bursts and prevent the steam, from the steam mains or other boilers adjacent or in connection with the disabled boiler, from coming back into the boiler. Its advantage can be easily seen when taking into consideration that power plant boiler header valves are usually some distance from the floor line and the boiler room would be full of steam, in case of a ruptured tube, before the attendants could, if it were possible, make their way to the header valves and close them by hand. This type of valve is also useful when a boiler is to be taken out of operation, it only being necessary to cease firing. As soon as the pressure falls below the line pressure, the valve automatically closes and cuts out the boiler. When firing is resumed and the steam pressure rises and equals the line pressure, the valve automatically puts the boiler on the line.

Some types of these boiler stop and check valves are made so that in addition to closing on a back flow of steam, they will also close in case a steam main should break beyond the outlet side of the valve. This additional feature is worthy of consideration when a plant has high pressure steam lines installed in tunnels or close quarters in buildings where workmen are liable to be trapped by escaping steam.

The use of these valves is not confined to boilers, as they may also be located on pipe lines supplying steam, air or other pressures. In addition to their automatic closing feature, they may be arranged to be opened and closed electrically from distant points by operating a push button, switch or other electrical or mechanical device.

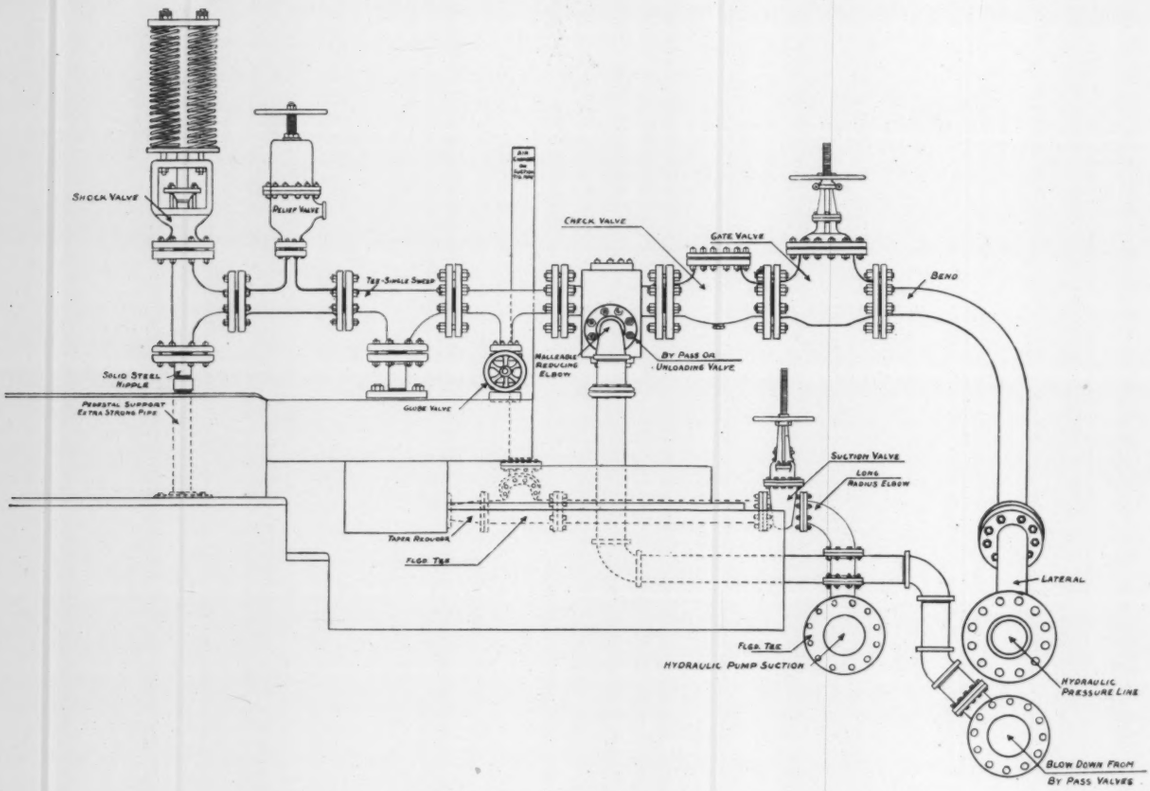


Fig 1. Check Valve as Used on Piping for Hydraulic Pumps

These valves have internal dash pots and should be inspected and cleaned periodically, although cases have been known where they have been in use for years with very little attention and, in an emergency, closed quickly, automatically and positively. They can be adjusted, according to local operating conditions, to close for any reasonable reduction in the outlet pressure (a pipe line in operation, bursting, will naturally cause a reduction in the pressure), usually, eight to ten pounds will be the required amount. This differential in pressure may have to be increased in some cases to prevent premature closing on account of a sudden demand for steam

when starting large engines, air compressors or turning on steam mains.

The Vacuum Breaker

A vacuum breaker is a type of check valve that will open inward and allow air from the atmosphere to pass through, into the tank reservoir or pipe line to which it may be attached, in case the interior pressure is suddenly relieved and falls below the atmospheric pressure. This prevents light-weight tanks and reservoirs from being crushed by the outside atmospheric pressure when the interior pressure is removed. It will also admit air to vent a closed tank when it is desired to empty the liquid contents rapidly. A swing check valve, installed with the inlet end open to the atmosphere, with a good tight fitting disc, rubber or leather faced for cold water or air pressures, makes a good vacuum breaker. One should be put on all light weight tanks, all large low pressure steam mains, all closed hydraulic equipment prefilling tanks, all closed feed water heaters, water stills, evaporators and similar vessels and equipment.

Placing of Check Valves

The installation of check valves is recommended for the following locations: Place a good check valve on the discharge line of every pump. If the pump is used in connection with a fire protection sprinkler system or other fire protection equipment, the type of check valve for the pump discharge and the foot valve for the pump suction, should be approved by the insurance underwriters before installing.

Check valves can be used to advantage on the discharge or outlet pipes of steam traps. Cases have been known of steam traps attached to large steam mains and discharging into return lines, handling water from many traps and other sources, having their water seal broken when the steam main was shut off. In the case of large heating mains, the same thing has happened when an unusual demand caused the pressure to drop below the atmospheric pressure and a partial vacuum was formed in the main. If a trap valve seat is leaking, it will cause water to be pulled back into the steam main. The steam pressure meeting a large amount of water is liable to cause serious trouble. A check valve on the discharge line of each and every steam trap would help to prevent this back flow of water. A swing check valve, and, as an added precaution, a vacuum breaker on the top of the steam main, would be satisfactory.

A check valve should be placed on the air supply connection to an overhead hoist. In case the air hose supplying this hoist should break, or the air supply line break, be shut off without warning, or anything unusual should happen to the compressed air supply, and the hoist were in operation with a suspended load, the check valve would close and prevent the load from coming down unexpectedly and endangering workmen. It would keep the air pressure in the hoist and not allow it to flow back through the air supply connection. In this case a ball check valve can be used and it should be placed as close to the hoist as possible.

Workmen in railway yards underneath cars working on the air brake equipment are in danger if the compressed air-supply is shut off without warning, the pipe line breaks, or an unusual demand for air lowers the pressure. Any of these would unexpectedly set the air brakes. As a protection to the workmen a check valve should be attached to the test hose used in connecting the cars to the air supply.

Hydraulic Press Installations

Hydraulic presses, of the overhead type where the upper platen moves downward on the working stroke, should have a check valve placed on the hydraulic

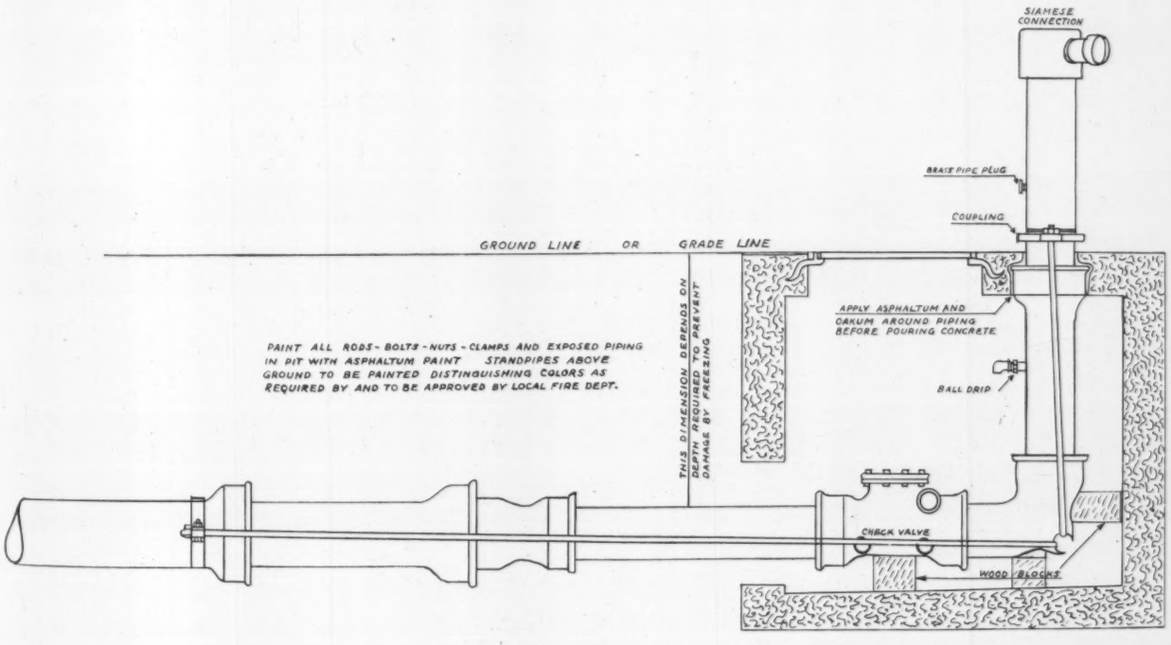

SIAMESE CONNECTION
BRASS PIPE PLUG
COUPLING
GROUND LINE OR GRADE LINE
APPLY ASPHALTUM AND OAKUM AROUND PIPING BEFORE POURING CONCRETE
PAINT ALL RODS - BOLTS - NUTS - CLAMPS AND EXPOSED PIPING IN PIT WITH ASPHALTUM PAINT STANDPIPES ABOVE GROUND TO BE PAINTED DISTINGUISHING COLORS AS REQUIRED BY AND TO BE APPROVED BY LOCAL FIRE DEPT.
THIS DIMENSION DEPENDS ON DEPTH REQUIRED TO PREVENT DAMAGE BY FREEZING
BALL DRIP
CHECK VALVE
WOOD BLOCKS

FIG. 2. UNDERGROUND INSTALLATION OF A CHECK VALVE FOR FIRE DEPARTMENT CONNECTION